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Analysis of convective heat transfer characteristics for a channel containing short multi-boards mounted with heat generating blocks

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Abstract

This study numerically investigates the two-dimensional forced convection in a channel containing short multi-boards mounted with heat generating blocks. The computation domain covers the all boards, and the region from the channel inlet to the location far away from the trailing edges of the boards. The flow and thermal characteristics are rigorously examined for the cases with various block height H_b, block spacing S_b, heat conductivity ratio of board to fluid K_{pf}, heat conductivity ratio of block to fluid K_{bf}, and Reynolds number Re. Results show that, owing to the effects of thermal interaction among the sub-streams of fluid through the conducting boards, the differences in transfer characteristics of the blocks mounted on different boards are rather significant. Comparing the maximum temperatures of the blocks on different boards, the maximum difference is up to 61% when $0 \leqslant K_{pf} \leqslant 200$, $100 \leqslant K_{bf} \leqslant 200$, $0.05 \leqslant H_b \leqslant 0.15$, $0.5 \leqslant S_b \leqslant 2.5$ and $200 \leqslant Re \leqslant 1000$. In addition the maximum deviation in the average Nusselt numbers $\overline{Nu}$ of the blocks on different boards is about 51%.

Keywords: Thermal interaction; Difference among boards

1. Introduction

Convective heat transfer from block heat sources to forced stream is widely encountered in engineering applications, such as in the cooling of electronic system, solar collectors, furnace and chemical processing equipments. Over the years, a lot of researchers have contributed their efforts to investigate the characteristics of forced convective flows over block heat sources. However, these reports mainly investigated the systems with blocks mounted on a single board. The articles on the system containing multi-boards mounted with block heat sources are not extensive. Besides most published work assumed that the sub-streams of fluid passing through each sub-channel formed by two adjacent boards had the same flow and thermal behaviors. Meanwhile the thermally-periodic boundary condition was imposed on two adjacent boards, i.e., only one single board with mounted blocks was investigated. Therefore these results did not illustrate the differences in the heat transfer characteristics for blocks mounted on different boards, which would be not appropriate for the system with a small number of boards. The main objective of this study is to rigorously examine the flow and thermal characteristics of each sub-stream in a channel containing short multi-boards mounted with heat generating blocks. The influences of thermal interactions among the sub-streams of fluid through the conducting boards on the heat transfer performance of blocks are conducted in detail. Great attention is focused on the differences in the heat transfer behaviors of the blocks mounted on different boards.

Because of its frequent occurrence in the industrial applications, the convective heat transfer from heated blocks have been studied by numerous researchers in the

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Nomenclature

D_a	dimensionless distance between the leading edge of board and the front surface of the first block, d_a/h_c
D_e	dimensionless distance between the channel inlet and the leading edge of board, d_e/h_c
D_f	dimensionless distance between the rear surface of last block and the trailing edge of board, d_f/h_c
H_b	dimensionless height of heat generating block, h_b/h_c
H_p	dimensionless thickness of board, h_p/h_c
h	heat transfer coefficient
K_{bf}	ratio of block to fluid thermal conductivities, k_b/k_f
K_{pf}	ratio of board to fluid thermal conductivities, k_p/k_f
n	outward normal direction to heat transfer surface
Nu	local Nusselt number, hh_c/k_f
P	dimensionless pressure, $p/(\rho_f u_e^2)$
Pr	Prandtl number, ν/α_f
$\dot{Q}$	heat input per unit width in each block
Re	Reynolds number, $u_e h_c/\nu$
S_b	dimensionless spacing between blocks, s_b/h_c
S_l	dimensionless spacing between the bottom channel wall and the low board, s_l/h_c
S_p	dimensionless spacing between boards, s_p/h_c
S_u	dimensionless spacing between the up channel wall and the up board, s_u/h_c
U	dimensionless longitudinal velocity, u/u_e
V	dimensionless transverse velocity, v/u_e
W_b	dimensionless length of heat generating block, w_b/h_c
X	dimensionless longitudinal coordinate, x/h_c
Y	dimensionless transverse coordinate, y/h_c

Greek symbols

α_f	thermal diffusivity of fluid
θ_b	dimensionless temperature of block, $(T_b - T_e)/(\dot{Q}/k_f)$
θ_f	dimensionless temperature of fluid, $(T_f - T_e)/(\dot{Q}/k_f)$
θ_p	dimensionless temperature of board, $(T_p - T_e)/(\dot{Q}/k_f)$
ν	kinematic viscosity
ρ_f	density of fluid

past two decades. Due to the space limitation, only a brief review of the previous literature is presented below. Kang et al. [1], and Nakayama and Park [2] experimentally examined the effects of block height and wall conduction on the heat convection for air flow over a two-dimensional, isolated block heat source mounted on plate. An experiment has been carried out by Nakamura and Igarashi [3] to investigate the forced convection heat transfer from a low-profile block placed in a rectangular duct. Kim et al. [4] examined the effects of flow pulsation on the characteristics of heat transfer from two heated blocks in a channel. Lehmann and Wirtz [5], and Agonafer and Moffatt [6] experimentally and numerically investigated the characteristics of flows over an array of two-dimensional, rectangular components mounted on channel wall. Heat was applied to the top surface of one component. It was found that the variation of heat transfer coefficient along the heated surface is rather different to that for smooth channel wall. Tsay et al. [7] conducted the performance of mixed convection in a horizontal duct with two heated blocks mounted on the bottom plate and baffle installed on the up plate. Sparrow et al. [8] presented an experiment on heat transfer and pressure drop for air flow in arrays of heat generating modules deployed along one wall of a flat rectangular duct by using the naphthalene sublimation technique. The effects of missing elements, height difference between modules and implanted barriers were investigated. Asako and Faghri [9,10] performed a three-dimensional analysis for laminar flow through an array of heated square blocks. Their results illustrated that the local heat flux on the top surface of block is higher than those on the front, side and rear surfaces. In addition, the Nusselt number decreases with the increasing block height and gradually approaches an asymptotic value. An experimental study on the convective heat transfer for water cooling of inline and staggered arrays of protruding elements has been reported by Garimella and Eibeck [11]. It was concluded that staggering the elements of the arrays could significantly enhance the heat transfer coefficient. Wong and Peck [12] performed an experiment to investigate the effect of altitude on the characteristic of heat transfer from blocks to air stream.

The articles [13–16] conducted the heat convection from a series of parallel plates with surface-mounted block heat sources. In these studies the periodic boundary conditions were adopted, and the transfer characteristics were investigated only for one of the boards. Besides, the boards were very long so that the effects of their leading and trailing edges could be neglected. Davalath and Bayazitoglu [13] numerically predicted the behaviors of forced convection between parallel plates mounted with two-dimensional multiple blocks. Results indicated that the heat flux distributions at the rear surfaces of blocks are much smaller than those at the front and top surfaces. An analysis of heat transfer from rectangular heated blocks mounted on vertical and horizontal multi-boards was carried out by Kim et al. [14]. They concluded that the impact of buoyancy effect on the flow and thermal fields is more pronounced for the vertically oriented channel. Furukawa and Yang [15] numerically investigated the thermal-fluid flow behaviors in a bundle of parallel boards with heat producing blocks. Kim and Anand [16] conducted the heat transfer between a series of parallel plates with surface-mounted discrete heat sources. The repeated boundary conditions were imposed both in the cross-stream and streamwise directions in the numerical simulation. The above-mentioned articles investigated the heat transfer characteristics for the block heat sources mounted on the plates or boards. On the other hand, Refs. [17–19] examined the thermal behaviors of heated components in channels. The components are not mounted on the plates or boards, thus all the surfaces of components expose to the cooling fluid. Chang and You [17] investigated the heat transfer from two hot blocks to mixed convective flow in a channel. Kundu et al. [18] presented the results for laminar flow over a row of in-line cylinders between two parallel plates. Yu et al. [19] analyzed the mixed convective characteristics for 3×3 array of horizontal cylinders in a vertical channel. Their results indicate that the mean Nusselt numbers for the cylinders in center line are substantially lower than those for the cylinders in the other lines.

The preceding review reveals that the information on the differences among the heat transfer characteristics of blocks mounted on different boards in a channel is rather limited. This study numerically investigates the convective heat transfer characteristics for a channel containing short multi-boards mounted with heat generating blocks. The computation domain includes the all boards, and the region from the channel inlet to the location far away from the trailing edges of the boards. The effects of thermal interaction among the sub-streams through the conducting boards on the heat transfer characteristics of blocks mounted on different boards are rigorously examined.

2. Analysis

The physical system under consideration, as illustrated in Fig. 1, is a two-dimensional plate channel containing three short boards with thickness h_p that are installed with the leading edge d_e downstream from the channel inlet. The low board is with s_l above the bottom channel plate, and the up board is with s_u below the up channel plate. The spacing between the boards is s_p. Each board is mounted with five heat generating blocks. The heat source is uniformly distributed in the volume of the blocks. A coolant stream enters the channel with uniform velocity u_e and temperature T_e. The stream divides into four sub-streams as it approaches the leading edge of boards. The sub-streams flow through the boards and remove the heat dissipated from the blocks. The flow characteristics might be different among the sub-streams owing to the effects of the leading and trailing edges of the boards, and the wall influence of the channel plates. In addition, the thermal interaction among the sub-streams through the conducting boards would affect the temperature distributions in the sub-streams. Consequently the heat transfer behaviors of the blocks mounted on the different boards might be different. This study rigorously investigates the characteristics of each sub-stream and the differences in the heat transfer performances of the blocks mounted on different boards.

The basic equations in dimensionless form describing the steady laminar forced convection in a two-dimensional plate channel containing multi-boards mounted with heat generating blocks are as follow.

For the region occupied by the fluid

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

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

$$U\frac{\partial V}{\partial X}+V\frac{\partial V}{\partial Y}=-\frac{\partial P}{\partial Y}+\frac{1}{Re}\left[\frac{\partial^2 V}{\partial X^2}+\frac{\partial^2 V}{\partial Y^2}\right] \tag{3}$$

$$U\frac{\partial \theta_f}{\partial X}+V\frac{\partial \theta_f}{\partial Y}=\frac{1}{RePr}\left[\frac{\partial^2 \theta_f}{\partial X^2}+\frac{\partial^2 \theta_f}{\partial Y^2}\right] \tag{4}$$

For the regions occupied by the blocks

$$\frac{\partial^2 \theta_b}{\partial X^2}+\frac{\partial^2 \theta_b}{\partial Y^2}+\frac{1}{H_b W_b K_{bf}}=0 \tag{5}$$

For the regions occupied by the boards

$$\frac{\partial^2 \theta_p}{\partial X^2}+\frac{\partial^2 \theta_p}{\partial Y^2}=0 \tag{6}$$

The governing equations are subjected to the following boundary conditions:

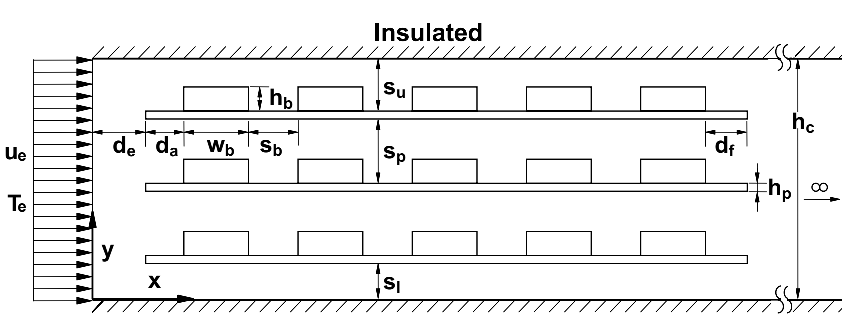

Fig. 1. Schematic diagram of the physical system.

At the channel inlet, $U = 1, \quad V = \theta_f = 0$ (7)

At the location far away from the trailing edges of the boards,

$$\frac{\partial U}{\partial X} = \frac{\partial V}{\partial X} = \frac{\partial \theta_f}{\partial X} = 0 \tag{8}$$

At the channel walls, $U = V = 0, \quad \frac{\partial \theta_f}{\partial Y} = 0$ (9)

At the fluid-block interfaces, $U = V = 0, \quad \theta_b = \theta_f, \quad K_{bf}\frac{\partial \theta_b}{\partial n} = \frac{\partial \theta_f}{\partial n}$ (10)

At the fluid-board interfaces, $U = V = 0, \quad \theta_p = \theta_f, \quad K_{pf}\frac{\partial \theta_p}{\partial n} = \frac{\partial \theta_f}{\partial n}$ (11)

At the block-board interfaces, $\theta_b = \theta_p, \quad \frac{K_{bf}}{K_{pf}}\frac{\partial \theta_b}{\partial Y} = \frac{\partial \theta_p}{\partial Y}$ (12)

Eqs. (7)–(12) refer to the usual no-slip conditions on all the solid walls, and the assumption of thermal insulation for channel walls. The stream is with uniform velocity and uniform temperature at the channel inlet, and with hydrodynamic and thermal fully developed conditions at the location far away from the trailing edges of boards. The continuities of temperature and heat flux are satisfactory at the fluid–solid interfaces, and at the block-board interfaces.

The local Nusselt number along the front, top and rear surfaces of a heat generating blocks is of interest to the thermal system designer. It is defined as

$$Nu = \frac{hh_c}{k_f} = -\frac{1}{\theta_f}\left(\frac{\partial \theta_f}{\partial n}\right)\text{(at the fluid-block interfaces)} \tag{13}$$

In addition, the average Nusselt number for a heat generating block is calculated by

$$\overline{Nu} = \frac{1}{L_h}\int_0^{L_h} Nu\,\mathrm{d}X_h \tag{14}$$

where X_h represents the dimensionless coordinate along the block surface and L_h is the total length of the front, top and rear sides of a block.

3. Solution method

The governing Eqs. (1)–(6) and boundary conditions (7)–(12) are solved by a numerical scheme derived from the SIMPLER algorithm. The finite difference equations for P, U, V and θ are obtained by integrating the respective continuity, X-momentum, Y-momentum and energy equations over the control volumes via a power-law scheme. To obtain enhanced accuracy, nonuniform grids are arranged in the computational domain. The grid density is higher in the vicinity of the boards and heat generating blocks to capture the drastic variations of the flow and thermal fields. The line-by-line method with iteration is employed to solve the systems of algebraic equations obtained from discretizing the pressure, momentum and energy equations. The solution is considered to be converged when the relative differences of U, V, and θ at each node between two consecutive iterations, as well as the overall energy imbalance of the system are less than a prescribed value of 10^{-4}.

The proposed numerical algorithm is validated in two ways. First, different numbers of grid lines in both the X- and Y-directions are employed to ensure that the solution is grid independent. The differences in U, V and θ at all grid points obtained from the 100×300 and 150×450 grid systems are less than 1% for a typical case with $Pr = 0.7$, $Re = 500$, $H_b = 0.15$, $W_b = 0.75$, $D_e = D_a = D_f = 1$, $S_b = 1.5$, $S_l = 0.1$, $S_u = S_p = 0.25$, $H_p = 0.05$, and $K_{bf} = K_{pf} = 100$. Therefore, the 100×300 grid system is chosen in the computation of the various cases to be presented. In addition, because of the elliptic nature of the present problem, it is necessary to make sure that the boundary conditions at $X \to \infty$ given by Eq. (8) do not artificially constrain the solution. To ensure that the results are not affected by the longitudinal size of the computation domain, tests are performed by varying the longitudinal size of the computation length downstream from the trailing edge of board L_c. The differences in the maximum temperatures of blocks between $L_c = 30$ and $L_c = 50$ are within 0.5%. Thus, in all the subsequent numerical simulation the computation domain with $L_c = 30$ is used to simulate the very long channel. Secondly, the results for the limiting situation with long, parallel boards and with cross-streamwise periodic conditions are compared to the relevant literatures. Good agreement is found between the present predictions and the results presented by Davalath and Bayazitoglu [13], and Kim et al. [14]. Through these program tests the proposed numerical scheme is considered to be appropriate for the present problem under investigation.

4. Results and discussion

Inspection of the forgoing analysis indicates that the flow and heat transfer characteristics in the present system depend on 14 parameters. Since a vast number of governing dimensionless parameters are required to characterize the system, a comprehensive analysis of all combinations of problems is not practical. The main objective here is to present a sample of results that illustrates the effects of thermal interactions among the sub-streams through the conducting boards on the heat transfer characteristics of heat generating blocks mounted on short boards in a channel. While computation can be performed with any combination, the effects of K_{pf}, K_{bf}, H_b, S_b and Re are investigated in this work. In particular, air ($Pr = 0.7$) flowing through the channel with $W_b = 0.75$, $D_e = D_a = D_f = 1$, $S_l = 0.1$, $S_u = S_p = 0.25$ and $H_p = 0.05$ is considered. The results are presented for the cases with K_{pf} varying from 0 to 200, K_{bf} from 100 to 200, H_b from 0.05 to 0.15, S_b from 0.5 to 2.5 and Re from 200 to 1000.

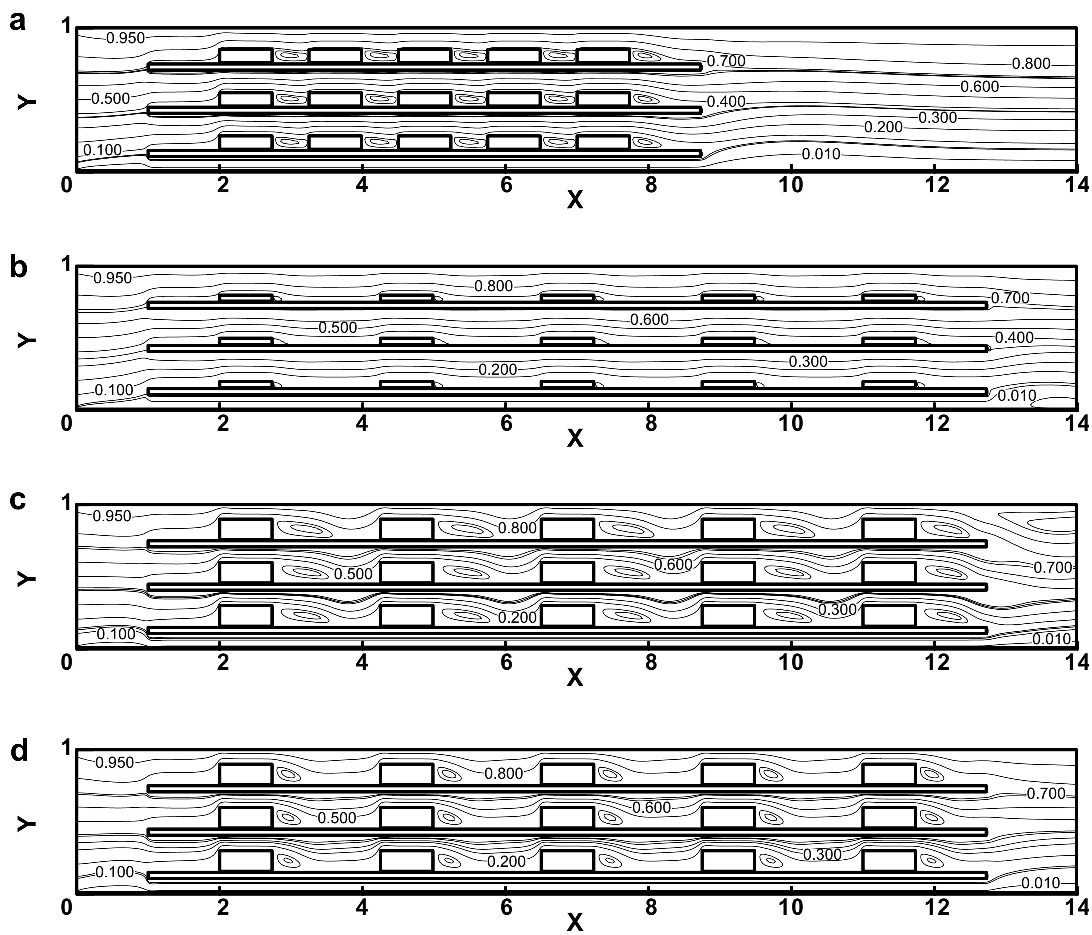

Fig. 2. The effects of dimensionless block spacing, block height and Reynolds number on the distributions of streamlines for $K_{bf} = K_{pf} = 100$, and (a) $Re = 500$, $H_b = 0.1$, $S_b = 0.5$; (b) $Re = 500$, $H_b = 0.05$, $S_b = 1.5$; (c) $Re = 500$, $H_b = 0.15$, $S_b = 1.5$; (d) $Re = 200$, $H_b = 0.15$, $S_b = 1.5$.

Initially, the flow structure in the channel is examined. The effects of the block spacing, block height and Reynolds number on the distributions of streamlines are portrayed in Fig. 2. It is observed in Fig. 2a that a primary recirculation cell appears in each gap between the blocks, and occupies nearly the whole gap. The results are rather similar to those reported by Davalath and Bayazitoglu [13], and Kim et al. [14]. It is interested to mention that the streamlines apparently distort in the region near the leading edges of boards when the flow approaches the boards. The distortion is more evident at the locations vicinity to the low board. The streamline with a value of 0.1 is distorted upward. For the case with small block height $H_b = 0.05$ and large distance between blocks $S_b = 1.5$, the results in Fig. 2b show that the recirculation cells are rather weak because the spacing between the blocks is much larger than the block height. For the case with large block height $H_b = 0.15$, the results in Fig. 2c indicate that the primary recirculation cells in the gaps among the blocks and behind the last blocks are much stronger than those plotted in Fig. 2b for $H_b = 0.05$. The streamlines near the bottom channel wall distort downward in front of the leading edge of low board, which is different to that plotted in Fig. 2b. In addition, a large recirculation zone is appeared in the region behind the up board. The results are different to those presented in Refs. [13], [14] which have the boards with very long length from the last block to the trailing edges of boards. Comparing the flow structures presented in Fig. 2c and d, it is found that the recirculation cells in the gaps is smaller for the case with lower Re. Meanwhile the secondary recirculation cell is not formed in the region behind the trailing edge of up board for the case with $Re = 200$. An overall inspection on Fig. 2a–d reveals that the channel wall, and the leading and trailing edges of the short boards can significantly affect the flow structures.

Fig. 3 illustrate the effects of K_{pf} and H_b on the temperature distributions for the system with fixed $K_{bf} = 100$, $S_b = 1.5$ and $Re = 500$. For the limiting case with insulated boards $K_{pf} = 0$, the results shown in Fig. 3a indicate that the isotherms of the boards and the respective blocks mounted on different boards are rather similar. The temperature distribution of each sub-stream flowing through the sub-channel between the boards, or between the up board and channel plate is also similar to one another. The hot spots in the blocks locate at block-board interfaces close to the rear surfaces of blocks. In addition, the

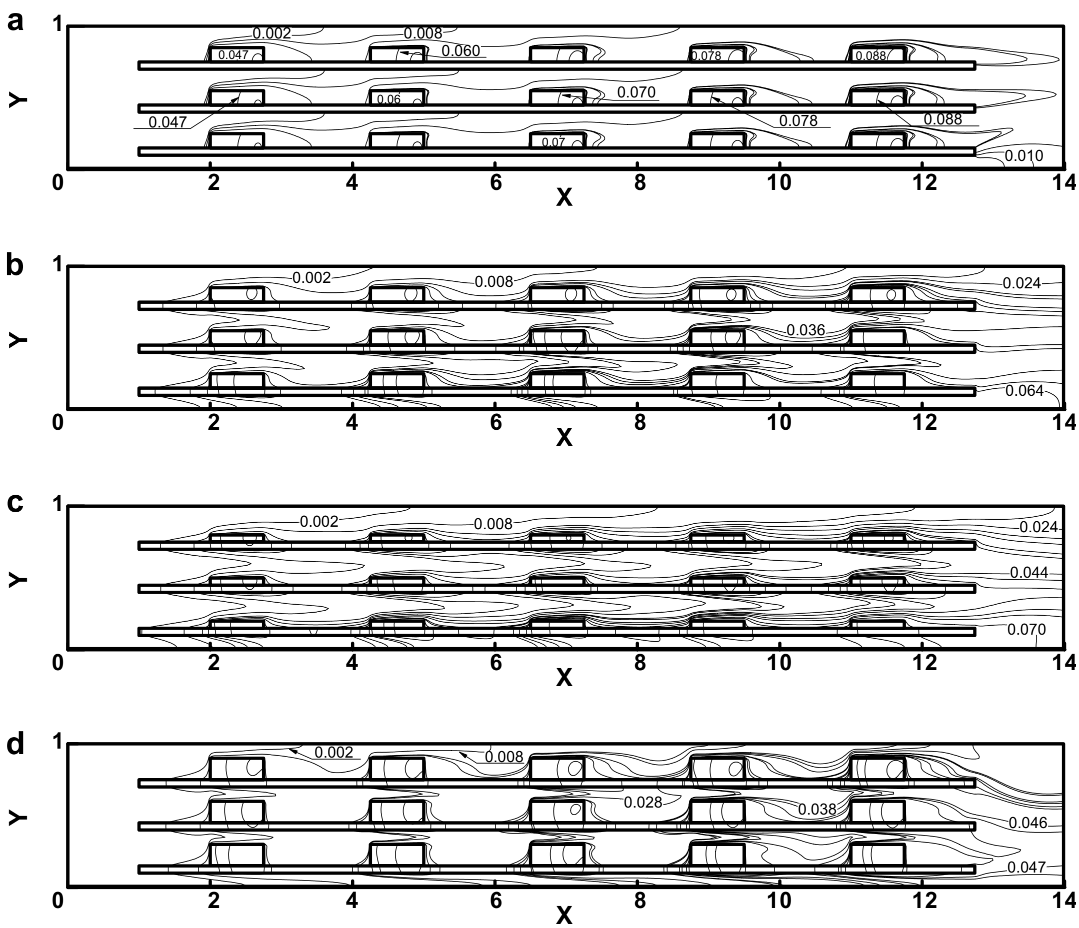

Fig. 3. The effects of ratio of board to fluid thermal conductivities, and dimensionless block height on the distributions of isotherms for $K_{bf} = 100$, $S_b = 1.5$, $Re = 500$ and (a) $K_{pf} = 0$, $H_b = 0.1$; (b) $K_{pf} = 100$, $H_b = 0.1$; (c) $K_{pf} = 100$, $H_b = 0.05$; (d) $K_{pf} = 100$, $H_b = 0.15$.

isotherms in the region near the rear surfaces of blocks are looser than those in the region near the front surfaces of blocks. These features are due to the fact that the recirculation cells in the gaps among the blocks and behind the last blocks are very close to the rear surfaces of blocks. The isotherms plotted in Fig. 3b reveals that, for the case with $K_{pf} = 100$, the thermal boundary layers along the up-side of one board and the back-side of the adjacent board merge at the locations upstream of the sub-channels among the boards. The temperatures in the low board and its blocks are significantly higher than that in the up board and its blocks. Also noted in this figure is that the hot spots of blocks are at the positions near the rear-top corner. These phenomena indicate that the K_{pf} may strongly affect the temperature distributions of the system. Furthermore the thermal interactions among the sub-streams of fluid through the conducting boards would be important in the developments of thermal field. It is important to mention in Fig. 3c that, even for a case with rather small block height $H_b = 0.05$, the thermal boundary layers along the adjacent boards can merge in the upstream regions of the sub-channels. A comparison on the isotherms plotted in Fig. 3b and c indicates that the merging locations of thermal boundary layers move toward upstream for the case with a higher H_b. Meanwhile the isotherms in Fig. 3d apparently distort downward in the region near the trailing edge of the up board. This is owing to the fact that the sub-stream turns to flow drastically downward in the region, as shown in Fig. 2c. The results represented in Fig. 3b–d clearly illustrate that the difference in temperature distributions among the sub-streams of fluid, and the blocks mounted on different boards are rather significant. Thus the treatment of thermally-periodic conditions adopted in the previous articles might be not practical in engineering applications when the system is with a small number of boards.

Attention is now turned to investigate the variations of local Nusselt numbers along the front, top and rear surfaces of the first to fifth blocks mounted on different boards. For the limiting case with insulated boards, the results in Fig. 4 show that the differences in the Nu's of blocks on different boards are negligible small. For a given block, the maximum heat transfer coefficient occurs at the front-top corner of block. The Nusselt numbers for rear surfaces are substantially smaller than those for front surfaces of blocks. This is due to the fact that the recirculation

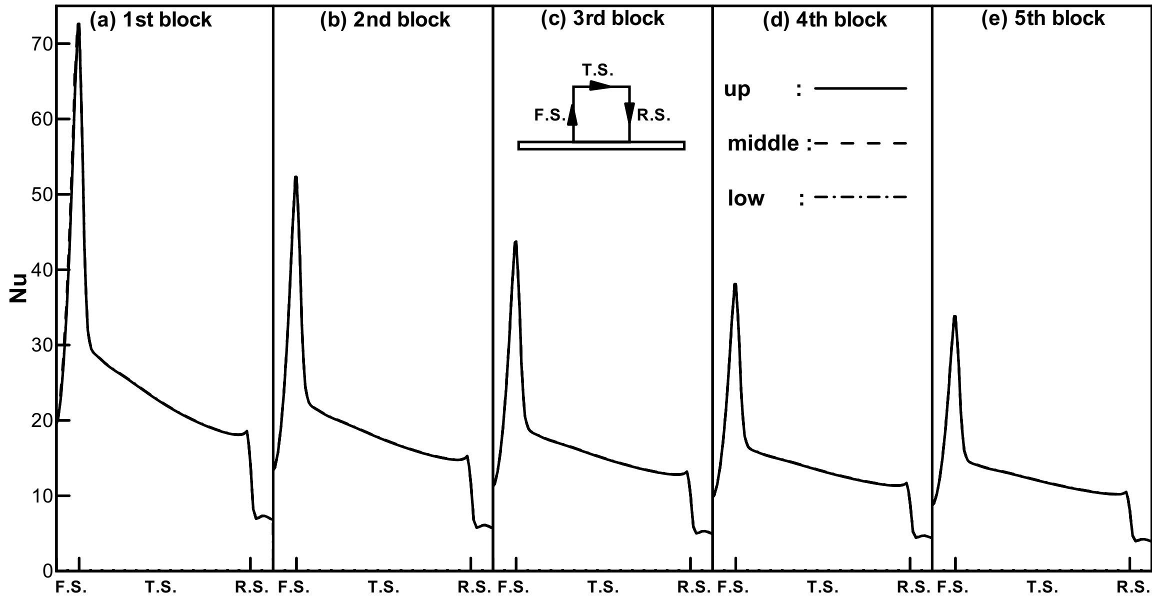

Fig. 4. The variations of local Nusselt numbers along the front, top and rear surfaces of heat generating blocks mounted on different boards for $K_{pf} = 0$, $K_{bf} = 100$, $H_b = 0.1$, $S_b = 1.5$ and $Re = 500$.

cells in the gaps between the blocks and behind the last blocks are very close to the rear surfaces of blocks. It is also mentioned that the *Nu*'s for the top surfaces slightly increase at the positions near the rear-top corners of blocks. This can be reasonably explained by the fact that the sub-streams turn to flow downward into the gaps when they pass the rear-top corners of blocks. For the case with conducting boards $K_{pf} = 100$, it can be clearly seen in Fig. 5 that the *Nu*'s for blocks mounted on the up board are significantly higher than those for the blocks mounted on the middle and low boards, except for the first blocks. However the *Nu*'s for blocks mounted on the middle and low boards are very close. A comparison on the results in Figs. 5 and 6 indicates that the differences in *Nu*'s of blocks on different boards become more evident for the case with larger H_b. It is interesting to note in Fig. 6 that the *Nu*'s for the top surfaces drop drastically with X in the region close to the front-top corners, then turn to increase until at the location about $0.2W_b$. This is due to the fact that, for the case with large H_b, the sub-streams separate from the top surfaces of blocks when they pass over the front-top corners of blocks. Then the sub-streams turn to flow

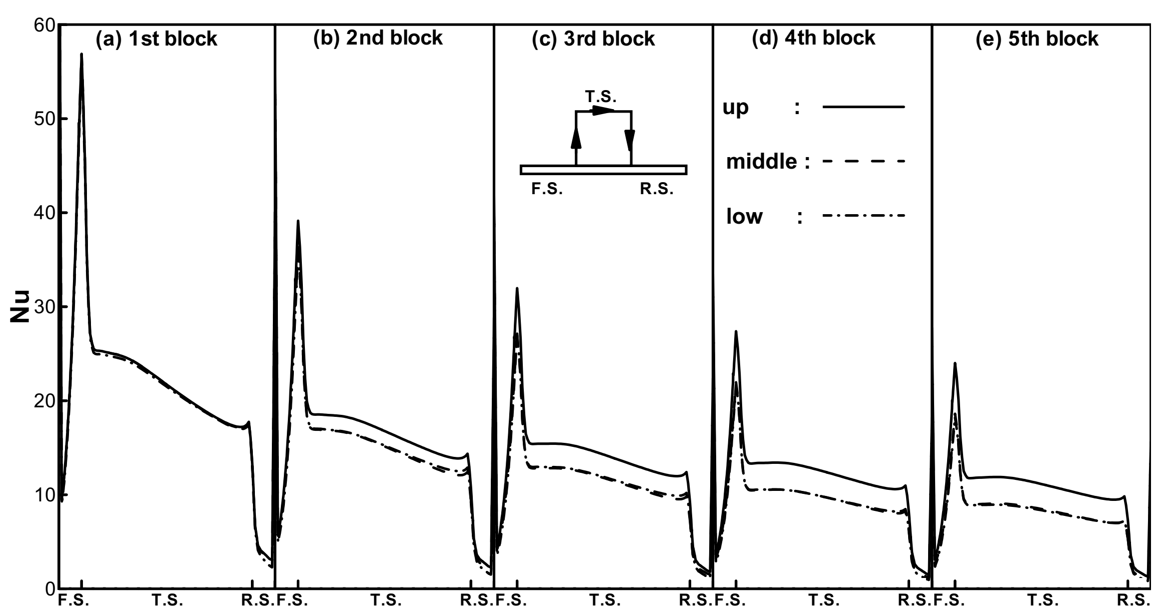

Fig. 5. The variations of local Nusselt numbers along the front, top and rear surfaces of heat generating blocks mounted on different boards for $K_{pf} = K_{bf} = 100$, $H_b = 0.1$, $S_b = 1.5$ and $Re = 500$.

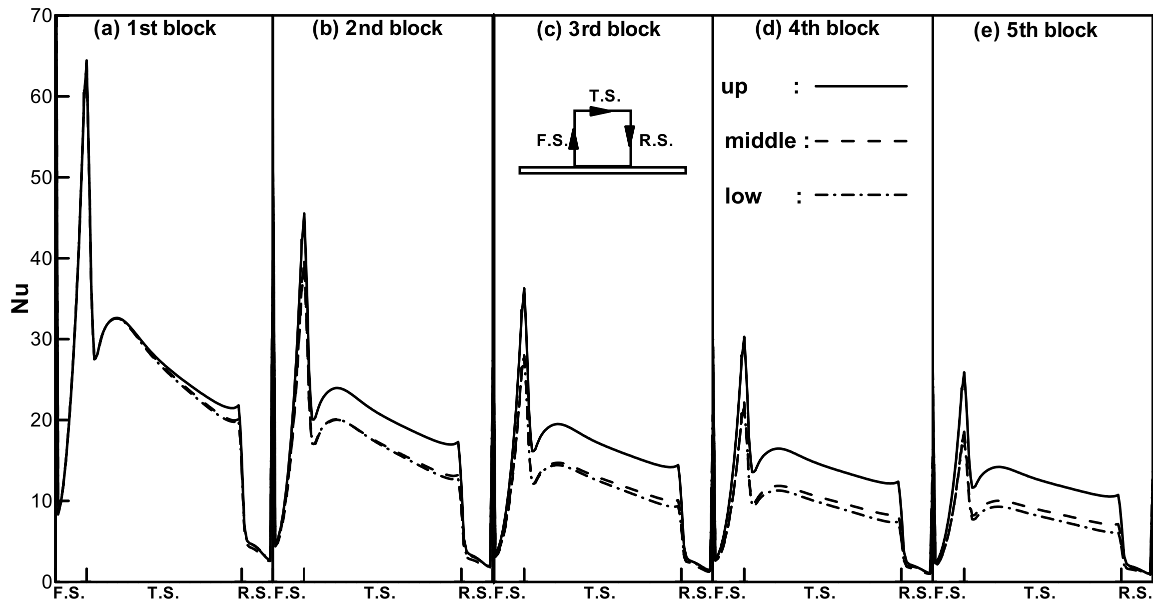

Fig. 6. The variations of local Nusselt numbers along the front, top and rear surfaces of heat generating blocks mounted on different boards for $K_{pf} = K_{bf} = 100$, $H_b = 0.15$, $S_b = 1.5$ and $Re = 500$.

Table 1
The comparison of the average Nusselt numbers for the last (fifth) blocks on the up, middle and low boards for the cases with various K_{pf} and K_{bf} when $Re = 500$, $H_b = 0.1$ and $S_b = 1.5$

K_{pf}	K_{bf}	$\overline{Nu}$			$\Delta\overline{Nu}$,%
		Up board	Middle board	Low board	
0	100	12.25	12.28	12.28	−0.2
100	100	10.41	7.95	7.77	33.9
200	200	9.74	7.30	7.20	35.2

$\Delta\overline{Nu} = (\overline{Nu}_{\text{up board}} - \overline{Nu}_{\text{low board}})/\overline{Nu}_{\text{low board}}$.

Table 2
The comparison of the average Nusselt numbers for the last (fifth) blocks on the up, middle and low boards for the cases with various H_b when $Re = 500$, $K_{pf} = K_{bf} = 100$ and $S_b = 1.5$

H_b	$\overline{Nu}$			$\Delta\overline{Nu}$
	Up board	Middle board	Low board	
0.05	9.69	7.76	7.43	30.4
0.1	10.41	7.95	7.77	33.9
0.15	10.86	7.73	7.20	50.8

$\Delta\overline{Nu} = (\overline{Nu}_{\text{up board}} - \overline{Nu}_{\text{low board}})/\overline{Nu}_{\text{low board}}$.

downward, and reattach to the top surfaces of blocks at the locations near $0.2W_b$. To more clearly indicates the differences in heat transfer characteristics of blocks mounted on different boards, Tables 1 and 2 list the average Nusselt numbers of last (fifth) blocks mounted on the up, middle and low boards. It can be read in Table 1 that, for the limiting case with insulated board $K_{pf} = 0$, the difference in $\overline{Nu}$'s of the blocks mounted on different boards is negligible. Meanwhile the $\overline{Nu}$ of the last (fifth) block mounted on the up board is significantly higher than those on the middle and low boards for the cases with conducting board. Also noted in Table 1 is that the $\overline{Nu}$'s for $K_{pf} = 0$ are higher than those for $K_{pf} = 100$ and 200. Table 2 represents the effects of H_b on the difference in $\overline{Nu}$'s of the fifth blocks mounted on the up, middle and low boards. The results show that the $\Delta\overline{Nu}$ is higher for the case with large H_b, and it is 50.8% when $H_b = 0.15$.

The hot spots in the system are important to the performance of thermal system. Finally the maximum temperatures in the last (fifth) blocks mounted on different boards are presented, and the effects of thermal interactions through the conducting boards on variations of the maximum temperatures are discussed. Table 3 presents the maximum temperatures in the last (fifth) blocks mounted on the up, middle and low boards for the cases with various K_{pf} and K_{bf}. For the limiting case with insulated boards $K_{pf} = 0$, the differences in θ_{max} among the different boards are negligible small. In addition, the θ_{max} for $K_{pf} = 0$ is substantially higher than that for $K_{pf} = 100$ and 200. For the cases with conducting boards, the differences in θ_{max} among the boards are rather significant. The $\Delta\theta_{max}$

Table 3
The comparison of the maximum temperatures in the last (fifth) blocks on the up, middle and low boards, and that obtained by adopting the thermally-periodic condition for the cases with various K_{pf} and K_{bf} when $Re = 500$, $H_b = 0.1$ and $S_b = 1.5$

K_{pf}	K_{bf}	θ_{max}				$\Delta\theta_{max}$, %
		Up board	Middle board	Low board	Periodic	
0	100	0.0901	0.0897	0.0897	0.1015	−0.4
100	100	0.0500	0.0597	0.0741	0.0681	48.2
200	200	0.0480	0.0580	0.0721	0.0659	50.2

$\Delta\theta_{max} = (\theta_{max,\text{low board}} - \theta_{max,\text{up board}})/\theta_{max,\text{up board}}$.

Table 4
The comparison of the maximum temperatures in the last (fifth) blocks on the up, middle and low boards, and that obtained by adopting the thermally-periodic condition for the cases with various H_b when $Re = 500$, $K_{pf} = K_{bf} = 100$ and $S_b = 1.5$

H_b	θ_{max}				$\Delta\theta_{max}$, %
	Up board	Middle board	Low board	Periodic	
0.05	0.0533	0.0633	0.0858	0.0719	61.0
0.1	0.0500	0.0597	0.0741	0.0681	48.2
0.15	0.0490	0.0564	0.0572	0.0640	16.7

$\Delta\theta_{max} = (\theta_{max,low\ board} - \theta_{max,up\ board})/\theta_{max,up\ board}$.

Table 5
The comparison of the maximum temperatures in the last (fifth) blocks on the up, middle and low boards, and that obtained by adopting the thermally-periodic condition for the cases with various S_b when $Re = 500$, $K_{pf} = K_{bf} = 100$ and $H_b = 0.1$

S_b	θ_{max}				$\Delta\theta_{max}$, %
	Up board	Middle board	Low board	Periodic	
0.5	0.0525	0.0605	0.0741	0.0696	41.1
1.5	0.0500	0.0597	0.0741	0.0681	48.2
2.5	0.0493	0.0596	0.0740	0.0679	50.1

$\Delta\theta_{max} = (\theta_{max,low\ board} - \theta_{max,up\ board})/\theta_{max,up\ board}$.

Table 6
The comparison of the maximum temperatures in the last (fifth) blocks on the up, middle and low boards, and that obtained by adopting the thermally-periodic condition for the cases with various Re when $K_{pf} = K_{bf} = 100$, $H_b = 0.1$ and $S_b = 1.5$

Re	θ_{max}				$\Delta\theta_{max}$, %
	Up board	Middle board	Low board	Periodic	
200	0.1018	0.1287	0.1548	0.1488	51.1
500	0.0500	0.0597	0.0741	0.0681	48.2
1000	0.0329	0.0365	0.0447	0.0408	35.9

$\Delta\theta_{max} = (\theta_{max,low\ board} - \theta_{max,up\ board})/\theta_{max,up\ board}$

is about 48% for $K_{pf} = K_{bf} = 100$, and 50% for $K_{pf} = K_{bf} = 200$. Tables 4 and 5 list the effects of H_b and S_b on the variations of θ_{max}. It is seen that the $\Delta\theta_{max}$ is higher for the cases with smaller H_b or longer S_b. The $\Delta\theta_{max}$ can be up to 61% when $H_b = 0.05$, as written in Table 4. The data listed in Table 6 show that $\Delta\theta_{max}$ is higher for the situation with smaller Re, and it is 51.1% for $Re = 200$. Tables 3–6 also list the θ_{max} obtained by adopting the thermally-periodic condition. It is interested to note that the values of θ_{max}'s for the situations with thermally-periodic condition are generally within the range of those for the boards. The above results illustrate that the thermal interactions through the conducting boards can cause significantly difference in the thermal characteristics of the blocks mounted on different boards.

5. Conclusions

This study aims to numerically investigate the 2D forced convection in a channel containing three short boards mounted with heat generating blocks. The flow and thermal characteristics have been examined for the cases with various K_{pf}, K_{bf}, H_b, S_b and Re. Owing to the effects of thermal interactions among the sub-streams of fluid through the conducting boards, the deviation in heat transfer characteristics of the blocks mounted on different board might be rather substantial. The deviation in temperature distribution is more significant for the cases with larger K_{pf}, K_{bf}, S_b or smaller H_b, Re. Comparing the maximum temperatures in blocks mounted on different boards, the maximum difference is up to 61% when $0 \leqslant K_{pf} \leqslant 200$, $100 \leqslant K_{bf} \leqslant 200$, $0.05 \leqslant H_b \leqslant 0.15, 0.5 \leqslant S_b \leqslant 2.5$ and $200 \leqslant Re \leqslant 1000$. Meanwhile the maximum difference in the average Nusselt numbers $\overline{Nu}$'s of the blocks mounted on the different boards is about 51%. These findings reveals that the thermally-periodic boundary condition imposed on two adjacent boards might result in substantial defect in engineering applications for the system with a small number of boards.

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