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Enhancement of mixed convection heat transfer in a three-dimensional horizontal channel flow by insertion of a moving block ☆

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ABSTRACT

Enhancement of mixed convection heat transfer rate of a heat surface in a three dimensional horizontal channel with insertion of a moving block is studied numerically. Bossinique assumption is not adopted, then methods of Roe scheme, preconditioning, and dual time stepping are needed to solve the governing equations. Contributions of important parameters of Gr/Re^2 and moving block velocity to the heat transfer rate are validated. Due to the consideration of natural convection, under situations of large magnitude of Gr/Re^2 a counter-effect for promotion of heat transfer rate is observed. Oppositely, under situations of small magnitude of Gr/Re^2, the enhancement of heat transfer rate is remarkably achieved.

1. Introduction

A problem of mixed convection of a heat surface in a horizontal channel flow is usually of practical importance, and widely considered in the design of devices such as heat exchangers, nuclear reactors, and solar energy systems. Thus, an effective method for improving the heat transfer performance of mixed convection in the horizontal channel flow is still expected urgently.

In addition to the development in convective heat transfer augmentation method detailedly reviewed by Bergles [1–2], well-known methods of the enlargement of heat dissipation surface [3–7], the disturbance of flow field [8–14] and the vibration of heat surface [15–18] are most usually adopted to enhance the forced convection heat transfer rate of the channel flow and the results of the above methods are remarkable and available.

However, due to the rapid formation of thermal boundary layer along the flow direction, the heat transfer rate of heat surface varies drastically from excellent to low level along the flow direction, and the amount of heat transfer rate in the upstream region of heat surface is multiple times of that in the downstream region of heat surface. As a result, a method of improving the heat transfer rate of downstream heat surface is expected to play an important role and to enhance the total heat transfer rate of heat surface in the channel flow. Fu et al. [19] proposed a method of installing a moving block tightly on a heat surface of a three dimensional channel. By way of mutual perpendicularity of the velocity directions of moving block and fluid flowing, the thermal boundary layer distributed on the heat surface is then destroyed by the reciprocating sweep of moving block and reformed by the inlet cooling fluid. From the results of Fu et al. [19], the reformation of thermal boundary layer will bring a remarkable contribution to the heat transfer mechanism. However, the forced convection is exclusively considered in the previous study [19]. From viewpoint of practical application, a situation of horizontal channel flow with a high temperature difference is usually involved, and it indispensably leads the interaction between the ascending fluid flowing caused by the high temperature difference and main fluid flowing caused by the inlet flow to occur and the heat transfer mechanism to become rather complicated. This subject is rarely investigated in the past.

Therefore, the aim of this study investigates the enhancement of mixed convection in a three dimensional horizontal channel flow by a moving block numerically. The moving block of which the movement is periodically transverse to the channel flow is installed along the channel flow. The immersed boundary method developed by Peskin [20] is adopted to solve the movement motion of moving block. The compressibility of fluid is considered to treat the natural convection induced by the high temperature difference situation, and other related methods of Roe scheme, preconditioning and dual time stepping are used to resolve the governing equations. The results show that under the situation of large magnitude of Gr/Re^2, the dominance of natural convection is strong and ascending fluids become remarkable. The inlet horizontal flow is then impeded and the effect of moving block on the heat transfer rate of heat surface is neutralized. As a result, the above situation has difficulty to promote the heat transfer rate of heat surface. Oppositely, under the situations of small magnitude of Gr/Re^2, the dominance of forced convection becomes apparent and the expectation of enhancement of heat transfer caused by the moving block is answered. Therefore, the above situations achieve the heat transfer rate of heat surface remarkably.

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Nomenclature

En	enhancement of the average heat transfer rate of the heat surface of a cycle in Eq. (11)
g	acceleration of gravity (9.802 m/s²)
Gr	Grashof number $\left(=\frac{g\beta(T_h-T_0)w_1^3}{\nu^2}\right)$
h	dimensional height of the slender block (m)
k	thermal conductivity (W/mK)
l_1	dimensional length of the channel (m)
l_2	dimensional length of the slender block (m)
l_3	the distance between the inlet and the heat surface (m)
l_4	the distance between the outlet and the heat surface (m)
Nu_Z	Nusselt number defined in Eq. (8-b)
Pr	Prandtl number
R	gas constant (287 J/kg/K)
Ra	Raleigh number($=$Gr $\cdot$ Pr)
Re	Reynolds number defined in Eq. (8-a)
t	time (s)
T	temperature (K)
T_0	temperature of surroundings (K)
T_f	film temperature (K)
T_h	temperature of heat surface (K)
w_b	moving velocity of the slender block (m/s)
W_b	dimensionless velocity of the slender block
w_1	dimensional width of the channel (m)
w_2	dimensional height of the channel (m)
w_3	dimensional width of the slender block (m)
x,y,z	Cartesian coordinates (m)

Greek symbols

θ	dimensionless temperature
μ	viscosity (N s/m^2)
μ_0	Surrounding viscosity (N s/m^2)
ρ	density (kg/m^3)
ρ_0	surrounding density (kg/m^3)

2. Physical model

The physical model investigated in this study is a three-dimensional horizontal channel and shown in Fig. 1. The direction of gravity g is vertical to the horizontal channel. The width, height, and length of the channel are w_1, w_2, and l_1, respectively. A heat surface ACDB of which the width, length, and temperature are w_1, l_2, and T_h, respectively, is installed on the bottom surface. The distances from the inlet and outlet to the heat surface are l_3 and l_4, respectively. Cooling fluids which possess uniform velocity u_0 and temperature T_0 flow into the channel. The thermal and flowing conditions of the cooling fluid at the outlet are fully developed. A moving block which has width w_3, length l_2, and height h is set on the heat surface. In order to avoid the impediment of channel flow in the regions near both vertical walls, the slender block moves at the constant speed of w_b from the point p_r to p_l back and forth, and the distances from the right and left walls to the points of p_r and p_l are the same and equal to $\frac{1}{4}w_1$. The gradient of pressure P on the all surfaces is zero. Except the heat surface, the other surfaces including the moving block are adiabatic.

The governing equations are expressed as follows.

$$\frac{\partial U}{\partial t}+\frac{\partial F_1}{\partial x}+\frac{\partial F_2}{\partial y}+\frac{\partial F_3}{\partial z}=S \tag{1}$$

The quantities included in U, F_i and S are separately shown in the following equations.

$$U=\begin{pmatrix}\rho\\ \rho u\\ \rho v\\ \rho w\\ \rho e\end{pmatrix} \tag{2}$$

$$F_i=\begin{pmatrix}\rho u_i\\ \rho u_i u_1+P\delta_{i1}-\mu A_{i1}\\ \rho u_i u_2+P\delta_{i2}-\mu A_{i2}\\ \rho u_i u_3+P\delta_{i3}-\mu A_{i3}\\ (\rho e+P)u_i-\mu A_{ij}u_j-k\dfrac{\partial T}{\partial x_i}\end{pmatrix},\ \forall i=1(x),2(y),3(z) \tag{3}$$

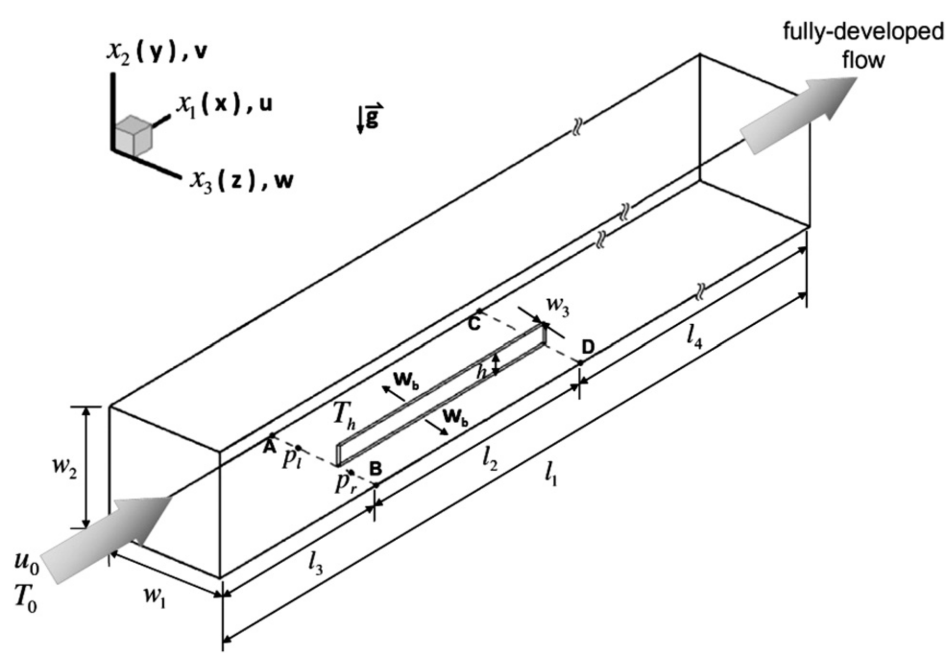

Fig. 1. Physical model.

where $A_{ij} = \frac{\partial u_i}{\partial x_i} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}(\nabla \cdot u)\delta_{ij}$ and the ideal gas equation is written by

$$P = \rho RT. \tag{4}$$

The Sutherland's law is adopted to evaluate the viscosity and the thermal conductivity as follows:

$$\mu(T) = \mu_0 \left(\frac{T}{T_0}\right)^{\frac{2}{3}} \frac{T_0 + 110}{T + 110} \tag{5}$$

$$k(T) = \frac{\mu(T)\gamma R}{(\gamma - 1)\mathrm{Pr}}$$

where $\rho_0 = 1.1842\ kg/m^3$, $\mu_0 = 1.85 \times 10^{-5}\ N \cdot s/m^2$, $T_0 = 298.0592K$, $\gamma = 1.4$, $R = 287\ J/kg/K$ and $\mathrm{Pr} = 0.72$,
and

$$S = \begin{bmatrix} 0 \\ 0 \\ -(\rho - \rho_0)g \\ 0 \\ -(\rho - \rho_0)gu_2 \end{bmatrix} \tag{6}$$

where $g = 9.81\,m/s^2$.

To simplify the analysis of results, the following dimensionless variables are made.

$$X = \frac{x}{w_1}, \quad Y = \frac{y}{w_1}, \quad Z = \frac{z}{w_1},$$
$$U = \frac{u}{u_0}, V = \frac{v}{u_0}, W = \frac{w}{u_0}, W_b = \frac{w_b}{u_0} \tag{7}$$

The compressibility and viscosity of the working fluid are considered, and the definition of the Reynolds number Re and the local Nusselt number Nu_Z on a certain YZ cross section of the heat surface is defined as follows.

$$\mathrm{Re} = \frac{\rho_0 u_0 w_1}{\mu_0} \tag{8-a}$$

$$Nu_Z = \frac{w_1}{k_0(T_h - T_0)}\left[k(T)\frac{\partial T}{\partial y}\right] \tag{8-b}$$

Since the conditions of the channel flow are dynamic and unsteady throughout, in Eq. (8-b) the temperature difference of $(T_h - T_f)$ which is usually used in a convection channel flow is conveniently substituted by $(T_h - T_0)$.

3. Numerical method

The numerical method adopted in this work is mainly modified from that used in the previous study [19] because of the addition of the gravity as a source term in the governing equations. Similar derivations of computation processes are indicated in Ref. [19].

4. Results and discussion

According to numerical tests, the optimal grid distribution of $200 \times 40 \times 40$ and related length variables are determined and shown as follows.

$$\begin{aligned} &w_2/w_1 = 1 \\ &w_3/w_1 = 0.05 \\ &h/w_1 = 0.5 \\ &l_1/w_1 = 15 \\ &l_2/w_1 = l_3/w_1 = 2.25 \\ &l_4/w_1 = 10.5 \end{aligned} \tag{9}$$

In Fig. 2, the results of velocities at the outlet of the channel are compared with the analytical solution [21] expressed in the following equation.

$$u(y,z) = \frac{4{w_1}^2}{\mu\pi^3}\left(-\frac{dp}{dx}\right)\sum_{i=1,3,5,\ldots}^{\infty}(-1)^{\frac{(i-1)}{2}}\left[1 - \frac{\cosh\left(\frac{i\pi z}{w_1}\right)}{\cosh\left(\frac{i\pi}{2w_1}\right)}\right] \times \frac{\cos\left(\frac{i\pi y}{w_1}\right)}{i^3} \tag{10}$$

In the central region a slight difference exists between both results, and in the other regions both results have good agreements.

In Fig. 3, the variations of the thermal fields of the channel caused by different Reynolds numbers are indicated. The larger the magnitude of Gr/Re^2 is, the dominance of natural convection becomes remarkable. Then the behavior of ascending heated fluids shown in Fig. 3(c) is more active than that shown in Fig. 3(a). However, this phenomenon is disadvantageous to the heat transfer rate of heat surface. Since the consideration of impediment of the ascending heated fluids mentioned

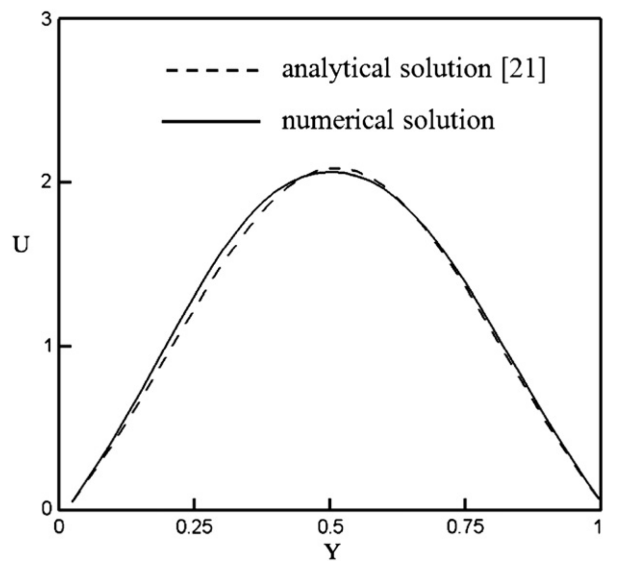

Fig. 2. Comparison of analytical and numerical solutions.

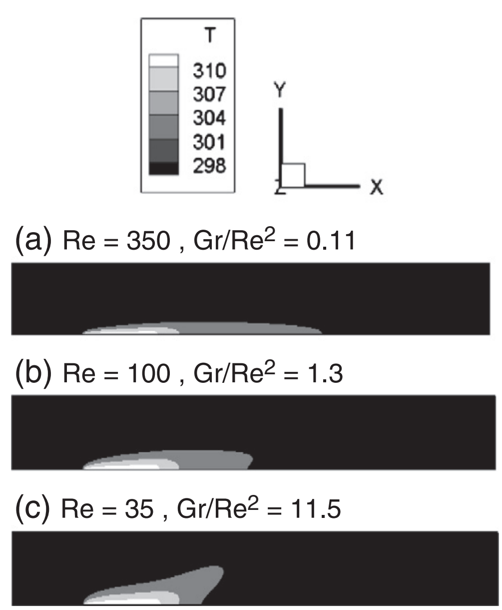

Fig. 3. Thermal fields at different magnitudes of Reynolds numbers.

above is necessary, the dimensionless height (h/w_1) of moving block is equal to 0.5.

In Fig. 4(a), the velocity vectors of fluids on the cross section of YZ plane at the leading edge $\overline{AB}$ of heat surface are indicated. The moving block moves from the right to left side and just passes through the central location. The velocities of fluids are induced by the movement of moving block and the directions of fluids are to the left below the height of moving block. A circulation zone is formed near the top of moving block, and above the height of moving block the directions of fluids are to the right in order to supplement the fluids below the height of moving block. The velocities of fluids attaching to the moving block are the same as the velocity of moving block for holding the no-slip condition. In Fig. 4(b), the isothermal lines corresponding to the condition of Fig. 4(a) are indicated. Since the location is at the leading edge of heat surface, the distribution of isothermal lines is dense near the heat surface region. Of those the distribution of isothermal lines in the right region of the moving block is denser than that in the opposite region. Because the movement of moving block is from the right to left, the thermal boundary layer in the right region of the moving block is reformed after the moving block passing through. This phenomenon is advantageous to the heat transfer rate of the heat surface. In Fig. 4(c), local Nusselt numbers distributed on the same location indicated in Fig. 4(a) and (b) are shown. Solid and dashed lines indicate the local Nusselt numbers of heat surface without and with the moving block, respectively. In the left region the fluids are pushed by the moving block, and in the right region the fluids supplement the vacant space and combine the inlet cooling fluids to share the reformation of thermal boundary layer. In both regions, the thermal boundary layers are drastically disturbed. Then the local Nusselt numbers of the situation with moving block are remarkably larger than those of the situation without moving block. The location $\overline{AB}$ is at the leading edge of heat surface; the cooling fluids are not heated yet. As a result, the enhancement of heat transfer rate achieved by the reformation effect occurring in the right region is prior to those achieved by the push effect occurring in the left region.

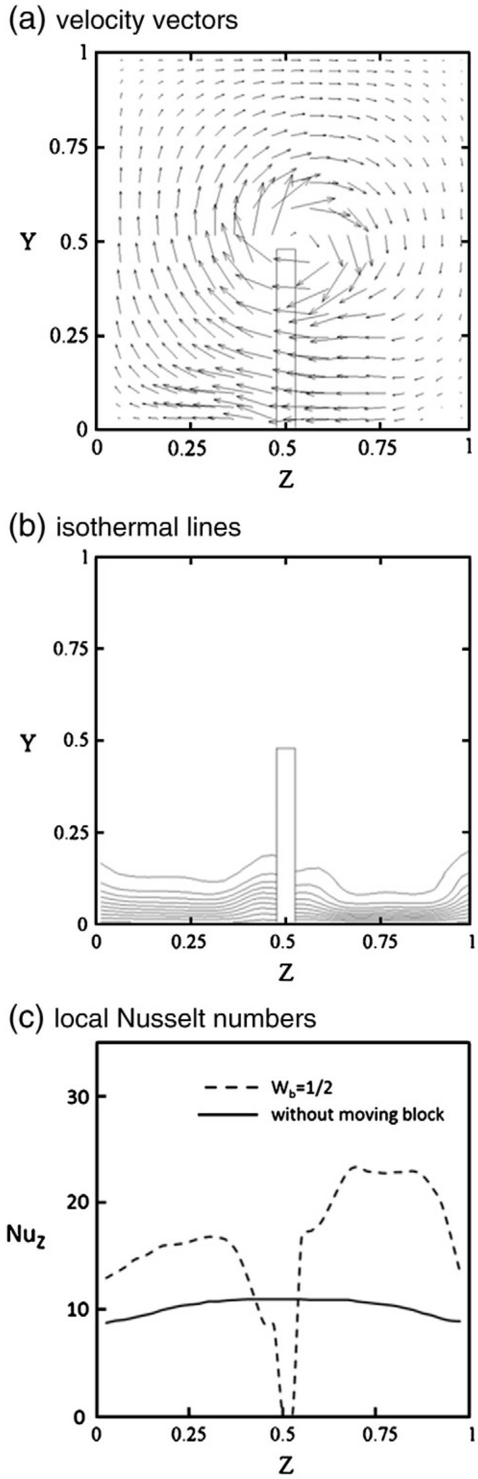

Fig. 4. Distributions of velocity vectors, isothermal lines and local Nusselt numbers on the cross section of YZ plane at AB location. (Re = 1000, Gr/Re^2 = 0.11, w_b = 0.5, ΔT_w = 100 K).

The velocity vectors of fluids on the cross section YZ plane at the location $\overline{CD}$ of heat surface are shown in Fig. 5(a). This location is the end of the moving block, and fluids are no longer confined in the region surrounded by the moving block and walls. Accompanying with the movement of moving block, the vectors of fluid velocities are then uniformly to the left under the height of moving block, and a circulation zone is observed near the top of moving block. At this location, in addition to the continuous growth of thermal boundary layer a mixed convection is also involved. The distribution of isothermal lines on this cross section shown in Fig. 5(b) is naturally sparser than those on the leading cross section shown in Fig. 4(b). In the left region the fluids from the upstream are only pushed by the moving block that causes the thermal boundary layer to be suppressed. In the right region, in the duration of reformation of thermal boundary layer the fluids from the upstream are mixed with the heated fluids from the left region caused by the circulation. As a result, the distribution of isothermal lines in the right region is no longer definitely denser than that in the left region. The corresponding local Nusselt numbers distributed are shown in Fig. 5 (c). Based on the phenomena mentioned above, the temperatures of fluids used to supplement the vacant space in the right region are raised that is disadvantageous to the heat transfer rate of heat surface. Consequently, the enhancement of heat transfer rate in the right region is inferior to that in the left region.

Enhancements of heat transfer rate of heat surface under different situations are tabulated in Table 1. The definition of enhancement (En) of heat transfer rate is defined as follows.

$$En = \frac{Nu_{with \quad moving \quad block} - Nu_{without \quad moving \quad block}}{Nu_{without \quad moving \quad block}} \tag{11}$$

Naturally, the smaller the magnitude of Gr/Re^2 is, the stronger the forced convection becomes. The ascending fluids caused by the natural convection are easily swept by the fast moving block velocity which is varied with the inlet fluid velocity and carried away by the inlet fluid flow. As a result, the enhancement of heat transfer rate is remarkable. However, in the situation of large magnitude of Gr/Re^2 the velocities of the inlet fluid flow and moving block become slow that results in the ascending fluids having difficulty to be swept and carried away by the moving block and inlet fluid flow, respectively,

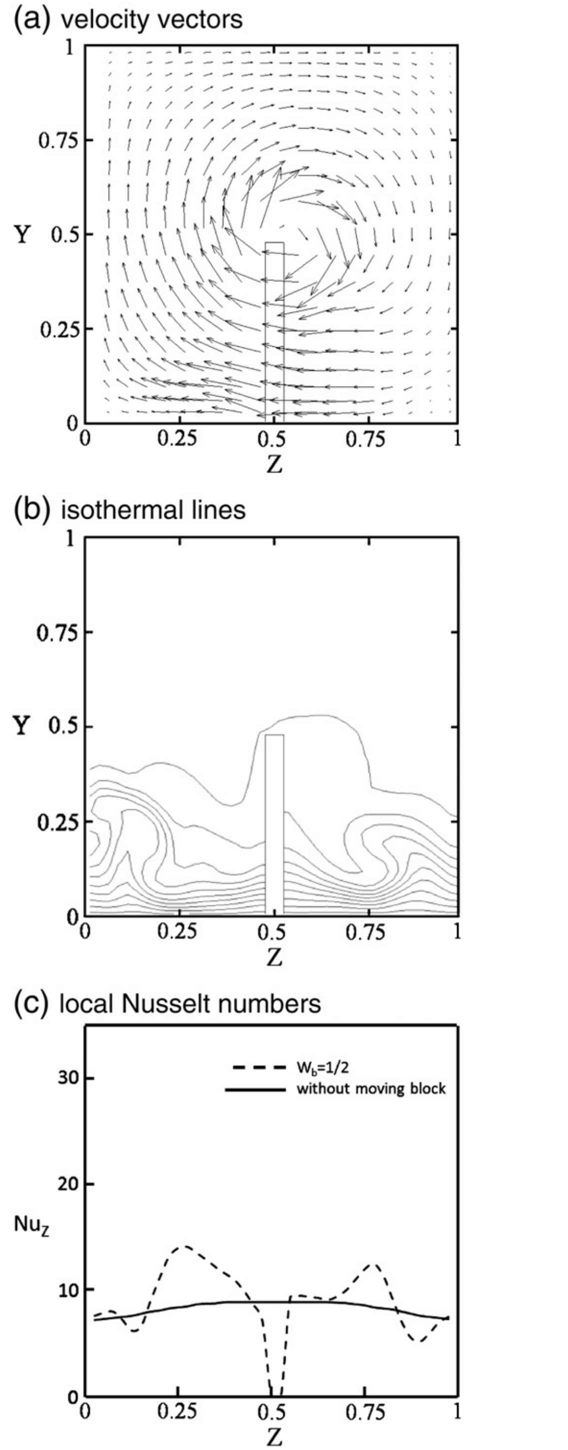

Fig. 5. Distributions of velocity vectors, isothermal lines and local Nusselt numbers on the cross section of YZ plane at CD location. (Re = 1000, Gr/Re^2 = 0.11, w_b = 0.5, ΔT_w = 100 K).

and being easily stagnant on the heat surface. Then a counter-effect of promotion of heat transfer rate is observed in these situations of large magnitude of Gr/Re^2.

In Table 2, the magnitudes of Gr/Re^2 are the same as those tabulated in Table 1, but the magnitudes of Reynolds numbers and Grashof numbers are decreased which means both effects of forced and natural convections are to be weakened. Ascending fluids are more difficultly carried away by the inlet flowing fluids which are interfered by the moving block. As a result, the enhancements are only achieved by a few situations.

Table 1
Enhancement of heat transfer rate of heat surface at different situations (ΔT_w ($T_h - T_o$) = 100 K).

Re	Gr/Re^2	V_b	$\overline{\mathrm{Nu}}$	En
1000	0.11	N/A	12.32	N/A
1000	0.11	0.25	14.32	16.3%
1000	0.11	0.5	17.04	38.3%
300	1.3	N/A	7.75	N/A
300	1.3	0.25	8.02	3.5%
300	1.3	0.5	8.56	10.5%
100	11.5	N/A	6.33	N/A
100	11.5	0.25	5.64	−11.0%
100	11.5	0.5	5.63	−11.0%

Table 2
Enhancement of heat transfer rate of heat surface at different situations (ΔT_w ($T_h - T_o$) = 12 K).

Re	Gr/Re^2	V_b	$\overline{\mathrm{Nu}}$	En
350	0.11	N/A	7.90	N/A
350	0.11	0.25	8.41	6.5%
350	0.11	0.5	8.80	11.5%
100	1.3	N/A	4.78	N/A
100	1.3	0.25	4.50	−5.8%
100	1.3	0.5	4.62	−3.3%
35	11.5	N/A	3.55	N/A
35	11.5	0.25	3.16	−11.0%
35	11.5	0.5	3.30	−7.0%

5. Conclusions

An investigation of enhancement of mixed convection heat transfer of a three dimensional horizontal channel flow by insertion of a moving block is studied numerically. Parameters of Gr/Re^{2}and moving block velocity are examined. Several conclusions are drawn.

1. Contributions of the movement of moving block to the heat transfer rate are validated under situations of small magnitude of Gr/Re^2. The maximum achievement is about 38%.
2. Due to the influence of natural convection, under situations of larger magnitude of Gr/Re^2 a counter-effect of promotion of heat transfer rate is observed instead.

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