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EFFECTS OF ECCENTRICITY OF CYLINDER AND BLOCKAGE RATIO ON HEAT TRANSFER BY AN OSCILLATING CYLINDER IN A CHANNEL FLOW

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ABSTRACT

A numerical simulation is performed to study the influence on heat transfer rate of heated walls in a channel with an oscillating cylinder. An arbitrary Lagrangian-Eulerian kinematic description method is adopted to describe the flow and thermal fields. A penalty consistent finite element formulation is applied to solve the governing equations. The effects of the eccentricity of cylinder ratio and blockage on the heat transfer characteristics of the heated wall are examined. The results show that the position and the diameter of the cylinder in the cylinder have great effects on the flow and thermal fields in the channel flow. © 2003 Elsevier Science Ltd

Introduction

A problem of force convection in a channel flow is of practical importance and widely considered in the design of devices such as heat exchangers, advanced gas-cooled reactor fuel elements and internal cooling passages of gas turbine. Therefore, the heat transfer phenomenon in a channel flow induced by the vortex shedding, which is caused by an oscillating cylinder, is also interesting and important in many engineering applications.

A summary of the literature on heat transfer in a laminar duct flow had been brought together in a book written by Shah and London [1]. It indicated analytical solutions for laminar fluid flow and force convection heat transfer in many passage geometries. Recently, numerous researches investigated the flow and thermal fields in a channel installed with ribs or stagnation blocks, such as Sparrow et al. [2], and Liou et al. [3-4], and the results showed that the ribs disturbed the flow field and enlarged the heat transfer area that caused the increment of the heat transfer rate. Lin and Hung [5] studied the transient forced convection heat transfer in a vertical rib-heated channel with a turbulence promoter, and found that the utilization of a turbulence promoter could effectively improve the heat transfer performance in the fully-developed region. As for a flow passing an oscillating cylinder, Cheng et al. [6,7], Karanth et al. [8], and Gau et al. [9] investigated heat transfer around a heated oscillating circular cylinder with experimental and numerical methods. The results found that the heat transfer increased remarkably as the flow approached the lock-in regime.

In existing studies, the investigation for the relationships with the oscillating cylinder set in the channel and the heat transfer of around walls of the channel is seldom. The heat transfer of the heated oscillating cylinder in a channel flow and the heat transfer of channel walls caused by an oscillating cylinder had been investigated by authors in the previous studies [10,11]. But, two important parameters of eccentric ratio and blockage ratio, which strongly affect the heat transfer of channel, are not investigated yet. The subject of the present work is therefore to investigate the influence on the heat transfer of the channel surface as the flow passing over the oscillating cylinder set in the channel under different eccentric and blockage ratio situations.

Physical Model

The physical model used in this study is shown in Fig. 1. A two-dimensional channel with height h_e and length w is used to simulate this problem. As the time $t > 0$, the cylinder is in oscillating motion normal to the inlet flow with amplitude l_c. The oscillating velocity of the cylinder is $v_c = 2\pi l_c \cos(2\pi f_c t)$. The behavior of the oscillating cylinder and the flow then affects mutually, and the variations of the flow field become time-dependent and are classified into a class of moving boundary problems. As a result, the ALE method is properly utilized to analyze this problem.

For facilitating the analysis, the following assumptions are made.

(1) The fluid is air and the flow field is two-dimensional, incompressible and laminar.

(2) The fluid properties are constant and the effect of the gravity is neglected.

(3) The no-slip condition is held on the interfaces between the fluid and cylinder.

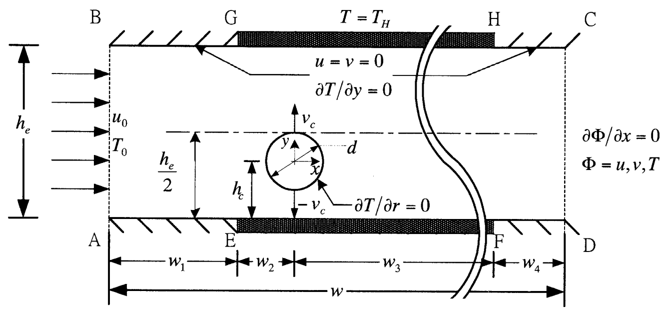

FIG 1
Physical model

Based upon the characteristics scales of w, u_0, ρu_0^2 and T_0, the dimensionless variables are defined as follows:

$$X = \frac{x}{h_e},\ Y = \frac{y}{h_e},\ U = \frac{u}{u_0},\ V = \frac{v}{u_0},\ \hat{V} = \frac{\hat{v}}{u_0},\ \tau = \frac{tu_0}{h_e},$$
$$V_c = \frac{v_c}{u_0},\quad L_c = \frac{l_c}{h_e},\quad H_c = \frac{h_c}{h_e},\quad F_c = \frac{f_c d}{u_0},\quad P = \frac{p - p_\infty}{\rho u_0^2}, \tag{1}$$
$$\theta = \frac{T - T_0}{T_H - T_0},\ \mathrm{Re} = \frac{u_0 h_e}{\upsilon},\ \mathrm{Pr} = \frac{\upsilon}{\alpha},\ BR = \frac{d}{h_e},\ ER = \frac{(h_e/2) - h_c}{(h_e/2)},$$

where $\hat{v}$ is the mesh velocity, v_c, f_c, h_c and l_c are the oscillating velocity, the oscillating frequency, the position and the oscillating amplitude of the cylinder, respectively. BR and ER are defined as eccentric ratio and blockage ratio, respectively.

According to the above assumptions and dimensionless variables, the dimensionless ALE governing equations are expressed as the following equations:

Continuity equation

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0, \tag{2}$$

Momentum equation

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

$$\frac{\partial V}{\partial \tau} + U\frac{\partial V}{\partial X} + \left(V - \hat{V}\right)\frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \frac{1}{\mathrm{Re}}\left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2}\right), \tag{4}$$

Energy equation

$$\frac{\partial \theta}{\partial \tau}+U\frac{\partial \theta}{\partial X}+\left(V-\hat{V}\right)\frac{\partial \theta}{\partial Y}=\frac{1}{\mathrm{Re}\,\mathrm{Pr}}\left(\frac{\partial^2 \theta}{\partial X^2}+\frac{\partial^2 \theta}{\partial Y^2}\right), \tag{5}$$

As the time $\tau > 0$, the boundary conditions are as follows:

On the inlet surface AB

$$U=1, \qquad V=0, \qquad \theta=0, \tag{6}$$

On the wall BG, HC, AE and FD

$$U=0, \qquad V=0, \qquad \frac{\partial \theta}{\partial Y}=0, \tag{7}$$

On the wall EF and GH

$$U=0, \qquad V=0, \qquad \theta=1, \tag{8}$$

On the outlet surface CD

$$\frac{\partial U}{\partial X}=0, \qquad \frac{\partial V}{\partial X}=0, \qquad \frac{\partial \theta}{\partial X}=0, \tag{9}$$

On the interfaces between the fluid and cylinder

$$U=0, \qquad V=V_c, \qquad \frac{\partial \theta}{\partial R}=0. \tag{10}$$

Numerical Method

The governing equations and boundary conditions are solved through the Galerkin finite element formulation and a backward scheme is adopted to deal with the time terms of the governing equations. The pressure is eliminated from the governing equations using the consistent penalty method. The velocity and temperature terms are expressed as quadrilateral element and eight-node quadratic Lagrangian interpolation function. The Newton-Raphson iteration algorithm is utilized to simplify the nonlinear terms in the momentum equations. The discretion processes of the governing equations are similar to the one used in Fu et al. [11].

Results and Discussion

The working fluid is air with $\mathrm{Pr} = 0.71$. The affects of main parameters of eccentric ratio ER, and blockage ratio BR on the heat transfer of heated walls are examined under $\mathrm{Re} = 500$ situation and the combinations of these parameters are tabulated in Table 1.

In the convection heat transfer of channel flow problem, it is necessary to define the local mean temperature T_{mx} of the stream as the environmental temperature.

$$T_{mx} = \frac{1}{\bar{u} \cdot h_e} \int_0^h uT \; dy \; ; \qquad \text{where } \bar{u} = \int_0^h u dy \tag{16}$$

The dimensionless variable θ_{mX} is defined as

$$\theta_{mX} = T_{mx} - T_0 / T_H - T_0 \tag{17}$$

The local Nusselt number is calculated by the following equation.

$$Nu_X = \frac{q_w''}{T_H - T_{mx}} \frac{h_e}{k} = \left(-\frac{\partial \theta}{\partial Y} \right) \frac{1}{1 - \theta_{mX}} \tag{18}$$

The average local Nusselt number from X_1 to X_2 along the heated surface is expressed as follows.

$$Nu_{(X_1)-(X_2)} = \frac{1}{|X_2 - X_1|} \int_{X_1}^{X_2} Nu_X dX \tag{19}$$

The time-averaged local Nusselt number per periodic cycle along the heated surface is defined by

$$\overline{Nu} = \frac{1}{\tau_p} \int_0^{\tau_p} Nu_X d\tau \qquad \text{where } \tau_p \text{ is time of a periodic cycle.} \tag{20}$$

The time-averaged Nusselt number of average local Nusselt number from X_1 to X_2 along the heated surface per periodic cycle along the heated surface is defined by

$$\overline{Nu_{(X_1)-(X_2)}} = \frac{1}{\tau_p} \int_0^{\tau_p} Nu_{(X_1)-(X_2)} d\tau \tag{21}$$

TABLE 1
Parameter combinations (Re=500)

	ER	*BR*
Case 1	0.0	0.25
Case 2	0.25	0.25
Case 3	0.5	0.25
Case 4	0.0	0.125
Case 5	0.0	0.5

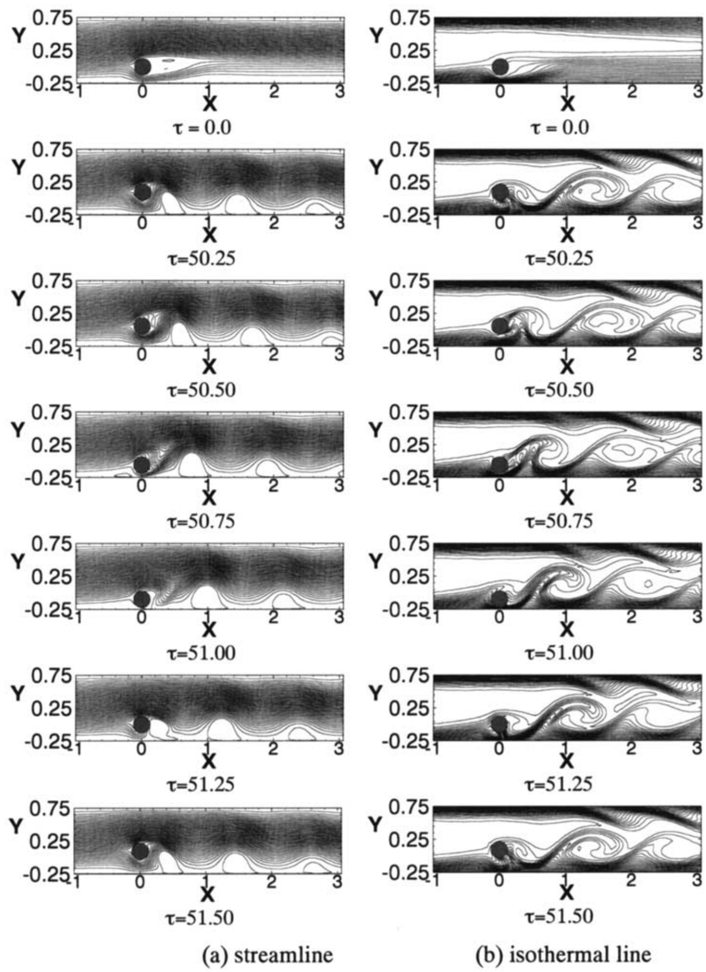

FIG 2
The variations of the streamlines and isothermal lines during one periodic cycle under case 3 situations

For clearly indicating the phenomena around the moving cylinder, the flow and thermal fields close to the oscillating cylinder are illustrated only in the following figures. The variations of the streamlines and isothermal lines under Re=500, L_c=0.1, F_c=0.2, $ER = 0.5$ and $BR = 0.25$ are indicated in Fig. 2a and Fig. 2b, respectively. At $\tau = 0$, the cylinder is set fixedly at $H_c = 0.25$ of the channel at steady state, the flow passing through the cylinder would be divided into two streams and accelerated. The flows depresses the thermal boundary of the wall near the cylinder, which cause the isothermal lines distributed right under the cylinder are denser than other regions. However, because of the flow resistance induced by the cylinder and the bottom wall, fewer fluids flow through the clearance between the cylinder and the bottom wall, which causes streamlines near the bottom wall behind the cylinder to be sparse. Therefore, the isothermal lines behind the cylinder near the bottom wall are sparser than those near the top wall. As time $\tau > 0$, the cylinder starts to oscillate with the velocity $V_c = 2\pi F_c L_c \cos(2\pi F_c \tau)$ and oscillating frequency $F_c = 0.2$. From $\tau = 50.25$ to $\tau = 51.50$, the variations of the streamlines during one periodic cycle of case 3 is shown. At $\tau = 50.25$, the cylinder moves upward and compresses the fluid upper the cylinder. Simultaneously, the clearance between the bottom wall and cylinder enlarges gradually and more fluids flow through this clearance. The cylinder turns downward after it reaches the maximum upper amplitude at $\tau = 50.50$. Since the moving direction of the cylinder is changed, the fluids near the top region of the cylinder replenish the vacant space induced by the movement of the cylinder, as shown from $\tau = 50.50$ to $\tau = 50.75$. After the cylinder reaches the lowest amplitude, the cylinder returns upward immediately and a new recirculation zone is formed behind the cylinder, shown at $\tau = 51.00$. As the cylinder moves upward, the recirculation zone grows up and the space between the cylinder and bottom wall enlarges gradually. For replenishing the vacant space induced by oscillating cylinder, more fluids flow through the space and push the recirculation zone formed before to shed from the cylinder and move downstream, shown at $\tau = 51.25$. Finally, the cylinder backs to the same position in the channel at $\tau = 50.25$ and completes a periodic cycle, shown at $\tau = 51.50$. As the cylinder oscillates in the channel, the vortex shedding and wavy-motion flow distrub both the thermal boundaries of the bottom and top walls intermittently, which causes the isothermal lines near both the walls to become denser than those of the cylinder being stationary in the channel. The streamlines and isothermal lines at the time $\tau = 50.25$ are identical with those at the time $\tau = 51.50$, which means that the variations of the flow and thermal fields become a periodic motion with time.

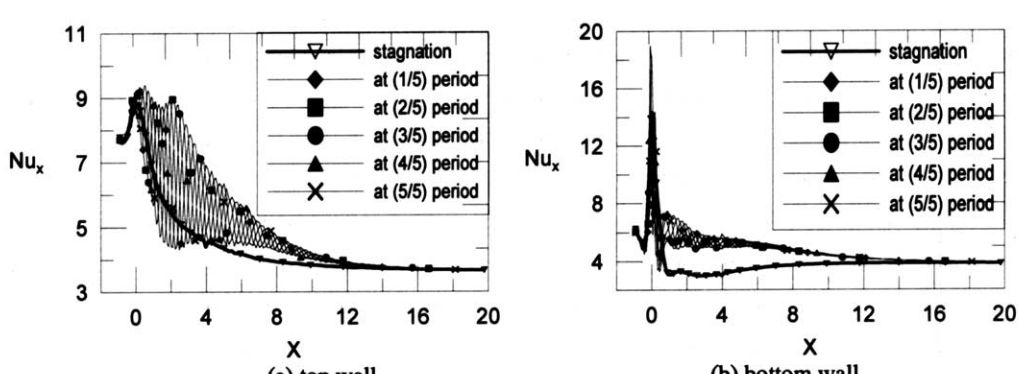

FIG 3
The distributions of average Nusselt number along the heated surface with time

Figure 3 shows the distributions of average Nusselt number along the top and bottom heated walls for case 3. The situations of the different times in the figure are corresponding to the those shown in the previous figures. Figure 3(a) shows the distributions of average Nusselt number along the top wall. Since the cylinder is far from the top wall, the influence of the oscillating cylinder on the top wall is slightly and the variation of heat transfer on the top wall relating to that of the stationary cylinder situation is not obvious. However, the fluids induced by the oscillating cylinder impinge the top wall behind the cylinder periodically, which causes the Nusselt number on the top wall behind the cylinder to vary violently with the oscillating period and promote the heat transfer rate obviously. Figure 3(b) shows the distributions of average Nusselt number along the bottom wall. The cylinder is very close to the bottom wall and the effect of the cylinder on the heat transfer of the bottom wall is larger than that of the top wall. Therefore, the variation of the Nusselt number on the bottom wall right under the oscillating cylinder is more violent. When the cylinder is stationary in the channel, the heat transfer rate of the bottom wall behind the cylinder decreases substantially due to the resistance of the cylinder. After the cylinder oscillates periodically, the vortex shedding disturbs the flow and thermal fields near the bottom wall. As a result, the heat transfer rate of the bottom wall behind the cylinder is enhanced effectively, as shown in Fig. 3(b).

Figure 4 shows the distributions of local Nusselt number Nu_x of the cylinder with time for different eccentric ratios in the channel under Re=500, L_c= 0.5 and F_c=0.2, situation. When the cylinder is close to the heated wall, the heat transfer rate of the heated wall under the cylinder is enhanced substantially, but the heat transfer rate of the region behind the cylinder is not promoted effectively. Contrarily, the heat transfer rate of the region behind the cylinder would be improved effectively as the cylinder is far away the heated wall; however, the increment of the Nusselt number of the heated wall under the cylinder is not as well as the cylinder closes to the heated wall.

The distributions of average Nusselt number along the heated surface with time for case 5 and case 4 situations are shown in Fig. 5(a) and 5(b), respectively. Because of the formation of recirculation zones near the walls, the heat transfer rate behind the cylinder is worse than that of the cylinder at the stationary situation. However, as the cylinder oscillates in the channel, the heat transfer rate behind the cylinder is promoted substantially, as shown in Fig. 5(a). When the cylinder becomes small, the influence of the oscillating cylinder on the heated wall would be decreased which causes the improvement of heat transfer to be slight and the effective region to be reduced, as shown in Fig. 5(b).

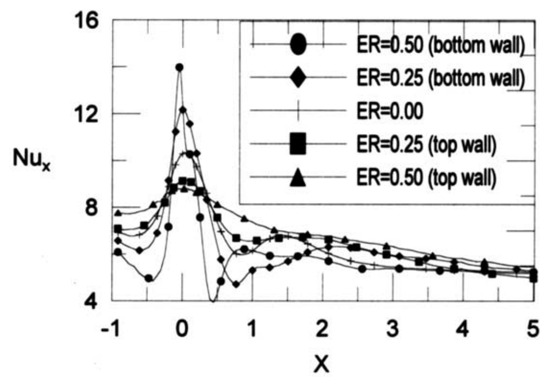

FIG 4
The variations of local Nusselt number along the heated surface for different eccentric under Re=500, L_c =0.1 situation and F_c =0.2 situation

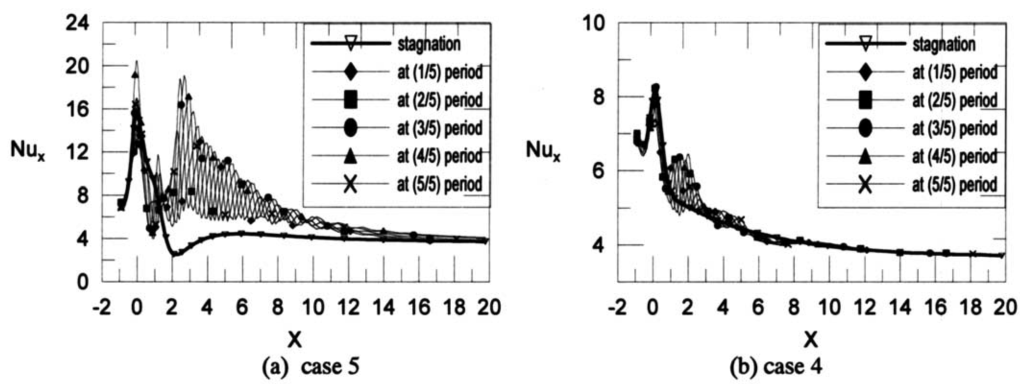

FIG 5
The distributions of average Nusselt number along the heated surface with time.

Figure 6 shows the distributions of local Nusselt number Nu_x of the cylinder with time for different blockage ratios of case 4, case 1 and case 5 under Re=500 and L_c = 0.1 situation, respectively. Doubtlessly, the larger the blockage ratio is, the velocity and disturbance of fluid are quicker and more drastic, respectively, and then the heat transfer rate is enhanced remarkably and the influence region is also enlarged.

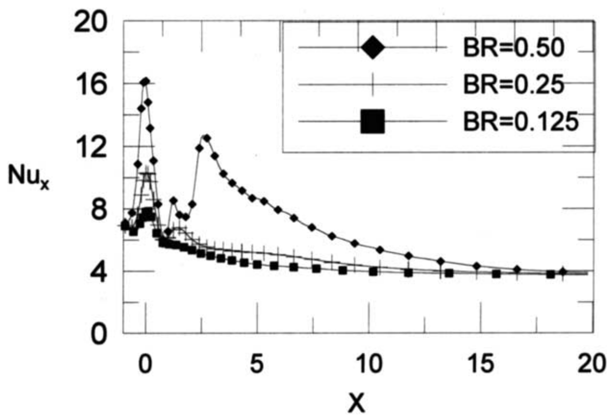

FIG 6
The variations of local Nusselt number along the heated surface for different blockage

Conclusions

The heat transfer characteristics of the heated wall in the channel with a transversely oscillating cylinder in a cross flow are investigated numerically. Some conclusions are summarized as follows:

1. The effects of insertion of an oscillating cylinder in the channel on heat transfer of wall are remarkably
2. An oscillating cylinder in the channel could make the flow to be wavy which improves the heat transfer rate of the channel wall.
3. When the cylinder closes to the heated wall, the heat transfer rate of the heated wall right under the cylinder is enhanced substantially. Contrarily, the heat transfer rate of the region behind the cylinder would be improved effectively as the cylinder is far away the heated wall.

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Nomenclature

BR	blockage ratio
ER	eccentric ratio
d	diameter of cylinder [m]
f_c	oscillating frequency [s^{-1}]
h_c	distance from bottom side to center of cylinder [m]
H_c	dimensionless distance from bottom side to center of cylinder
h_e	height of the channel [m]
k	thermal conductivity
l_c	oscillating amplitude [m]
L_c	dimensionless oscillating amplitude
$\overline{Nu}$	periodic-averaged Nusselt number
Nu_X	local Nusselt number
$Nu_{(X_1)-(X_2)}$	average Nusselt number from X_1 to X_2
p	dimensional pressure [$N \cdot m^{-2}$]
p_∞	reference pressure [$N \cdot m^{-2}$]
P	dimensionless pressure
q_w''	heat transfer rate from heated wall
Pr	Prandtl number
r	radial coordinate [m]
R	dimensionless radial coordinate
Re	Reynolds number
t	time [s]
T	temperature [K]
T_H	temperature of heated region [K]
T_{mx}	local mean temperature [K]
T_0	temperature of inlet fluid [K]
u, v	dimensional velocities [$m \cdot s^{-1}$]
U,V	dimensionless velocities
u_0	velocities of inlet fluid [$m \cdot s^{-1}$]
v_c	oscillating velocity [$m \cdot s^{-1}$]
V_c	dimensionless oscillating velocity
$\hat{v}$	mesh velocity [$m \cdot s^{-1}$]
$\hat{V}$	dimensionless mesh velocity
w	length of channel [m]
w_1	length of adiabatic region before cylinder [m]
w_2	length of heated region before cylinder [m]
w_3	length of heated region behind the cylinder [m]
w_4	dimensional length of the adiabatic region behind the cylinder [m]
x, y	dimensional Cartesian coordinates [m]
X,Y	dimensionless Cartesian coordinates

GREEK SYMBOLS

Φ	computational variables
ν	kinematic viscosity [$m^2 \cdot s^{-1}$]
θ	dimensionless temperature
θ_{mX}	dimensionless local mean temperature
ρ	density [$kg \cdot m^{-3}$]
τ	dimensionless time

OTHER

$\| \|$	absolute value

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