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Heat transfer enhancement by using metallic filament insert in channel flow

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Abstract

To increase the uniformity of radial temperature profile for fluid flow in square channel and consequently to enhance the heat transfer, the use of filament insert is developed and investigated experimentally and numerically. The metallic filament of very large length-to-diameter ratio installed in the radial direction, without contacting the channel wall, will alter the radial temperature profile. The numerical results for friction factor and Nusselt number are presented for a range of Reynolds number from 200 to 1200. Both numerical and experimental results show that the convective heat transfer is considerably enhanced by the metallic filament insert. It is more important that the values of performance evaluation criterion are raised to 3–8. This implies that the advantage of present approach lies in the less additional pressure drop while heat transfer is enhanced.

Keywords: Heat transfer enhancement; In-channel flow; Metallic filament insert

1. Introduction

In the development of modern industrial world, there is always a pressure to reduce the capital and running costs for heat transfer process. The need for high performance thermal systems has stimulated research interest in augmentation of heat transfer. This can be stated to have the encouragement and accommodation of high heat fluxes [1]. Tremendous works on heat transfer enhancement have been conducted and a large number of techniques for convective heat transfer enhancement have been developed since 1970s. These techniques are usually segregated into two groups: passive and active techniques [2–4].

Passive techniques cover the use of extended surfaces, surface roughness and inserts, etc., while surface vibration, fluid blow or suction, and applying electric or magnetic field are named active techniques, which require external power supply. The mechanism of conventional passive techniques can be attributed to the increase of the heat transfer area and/or convective heat transfer coefficient. One of the following ways will result in the rise of the heat transfer coefficient: (a) mixing the main flow and/or the flow in the near-wall region by using rough surface, inserts, etc.; (b) reducing the flow boundary layer thickness by using offset strip fins, jet impingement, etc.; (c) creating the rotating and/or the secondary flow by using swirl flow device, duct rotation, etc.; (d) raising the turbulence intensity by using surface roughness, turbulence promoter, etc. The main problem for the existing techniques lies in the large additional pressure drop associated with heat transfer enhancement. This is why many highly efficient techniques of heat transfer enhancement have no or less practical applications.

Based on an analog between heat convection and heat conduction, Guo et al. [5] studied the mechanism of convective heat transfer from a second look and proposed novel approaches to enhance convective heat transfer. These novel approaches involve improving the uniformity of velocity and temperature profiles as well as increasing the included angle between dimensionless velocity and temperature gradient vectors. In the present study, filament insert in the channel is developed to improve directly the uniformity of temperature profile and consequently to enhance convective heat transfer. Both numerical and experimental studies are performed

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Nomenclature

d_e	hydraulic diameter of filament
D_e	hydraulic diameter of the channel
f	friction factor
h_b	average heat transfer coefficient
L	length of the channel
Nu_b	overall Nusselt number of filament insert channel
Nu_0	Nusselt number of smooth channel
p	pressure
PEC	performance evaluation criteria
Pr	Prandtl number
Q	heat transfer rate per module
Re	Reynolds number
S	pitch of the filaments
T	temperature
T_w	wall temperature
u_b	cross-sectional average of u
u, v, w	velocity components along x-, y- and z-coordinates
x, y, z	x-, y- and z-coordinates
ΔT_m	logarithmic mean temperature difference

Greek symbols

δ	space between filament tip and channel wall
λ_m	thermal conductivity of filament material
μ	dynamic viscosity of the fluid
ρ	density of the fluid

to examine the effect of filaments on heat transfer in a laminar channel flow.

2. Increasing uniformity of temperature profile with filament insert

As mentioned previously, more additional pressure drop will be produced to increase heat transfer rate by traditional techniques. This is because heat transfer is enhanced through the variation of velocity field and the consequent increase of temperature gradient at the wall surface. The idea in this paper aims at directly improving the uniformity of temperature profile without or with less change of velocity field. A kind of insert composed of metallic filament is proposed and investigated to enhance heat transfer with less additional pressure drop. Fig. 1 illustrates a filament insert channel and defines its geometric variables. Filament insert is an array of very large length-to-diameter filaments made of high-conductivity material. They are distributed sparsely in the channel, vertical to each other, and perpendicular to the flow direction and wall surface. The tips of filament are not contacted to the channel wall, with distance δ. Thus, the presence of filament insert does not have direct contribution to heat transfer between fluid and wall. Its enhancement effect lies in the altered temperature field in its nearby region. The temperature profile after fluid flowing over the high-conductivity filament becomes more uniform and temperature gradient near channel wall is elevated. From the view of increasing uniformity of temperature profile in the radial direction, the filaments should be installed radially. Another feature of filament insert is its rather small hydraulic diameter and high volume porosity (ratio of volume not occupied by filaments) which can effectively reduce the additional pressure drop.

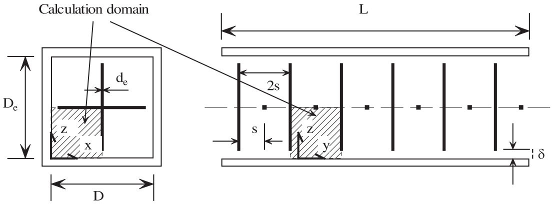

Fig. 1. Geometry of the problem.

3. Statement of computation problem

In the configuration shown in Fig. 1, the cross-section of channel and filament under numerical investigation is a square with hydraulic diameter of D_e and d_e, respectively. Filaments are placed uniformly at a pitch $2S$ in the square channel. The flow attains a periodic fully developed character after the initial entrance region, so that the flow field repeats itself in an identical fashion in successive geometrical modules formed by the presence of the filaments. The numerical method [6] was used to compute the flow and heat transfer characteristics of a typical module shown in Fig. 1. Generally, the results for the typical module in the periodic fully developed region are usually sufficient for the whole channel, since for channels containing a large number of filaments, the effect of entrance region becomes unimportant.

The present study provides 3-D numerical solutions for a typical module. The governing equations are written as:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0, \tag{1}$$

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right) = -\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right), \quad (2)$$

$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right) = -\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right), \quad (3)$$

$$\rho\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right) = -\frac{\partial p}{\partial z}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right), \quad (4)$$

$$\rho c_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}+w\frac{\partial T}{\partial z}\right) = \lambda\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right). \quad (5)$$

In the current periodic fully developed flow, the fluid properties are regarded as constant and constant wall temperature boundary condition ($T_w = 100^\circ$C) is employed. The numerical solution is obtained using a grid with 40 subdivisions in the y direction and 20 subdivisions in the x and z directions. The density of points is increased near the channel wall, filament side, and filament tip due to steep gradients at these locations. Computations are performed for different Reynolds numbers and for different geometrical parameters δ and S, in which d_e/D_e is fixed at 0.05. The heat transfer results are obtained for two values of fluid Prandtl number, namely $Pr = 0.7$ for air and $Pr = 4$ for water, and three values of filament material heat conductivity, namely $\lambda_m = 4, 40$, and 400 W/(m K), corresponding to wood, steel and copper, respectively.

The present numerical simulation provides extensive information in terms of the fields of velocity, pressure and temperature. Also, local and overall heat transfer coefficients have been calculated. Here, only flow and overall heat transfer characteristics are presented and discussed. $fRe/(fRe)_0$ and Nu_b/Nu_0 will be used to represent the flow and heat transfer characteristics, respectively. It is well known that for a fully developed laminar flow in a square channel, $(fRe)_0 = 56.9$, and $Nu_0 = 2.98$ under constant wall temperature condition. Friction factor, Reynolds number and overall Nusselt number are given by:

$$f = -(\mathrm{d}p/\mathrm{d}y)D/[(1/2)/\rho u_b^2], \quad (6)$$

$$Re = \rho u_b/\mu, \quad (7)$$

$$Nu_b = h_b D/\lambda_f, \quad (8)$$

where the mean heat transfer coefficient is calculated from

$$h_b = Q/[4D \times 2S \times \Delta T_m], \quad (9)$$

where Q is the heat transfer rate per module, $4D \times 2S$ is the heat transfer area of channel wall, and ΔT_m is the logarithmic mean temperature difference. It should be noted that the area used in the definition of h_b does not include the extra area provided by the filaments, which do not have direct contribution to heat transport. Thus, h_b is based on the nominal area of a corresponding smooth channel. Meanwhile, to value the heat transfer enhancement for a given pumping power, a performance evaluation criteria defined by PEC $= (Nu_b/Nu_0)/(f/f_0)^{1/3}$ [7] is employed.

4. Results and discussion

The temperature profile of fluid is plotted in Fig. 2. It can be seen that the dimensionless temperature profile of fluid flowing over the filament becomes more uniform and the temperature drop falls in the region near the channel wall.

The results for all the cases studied are plotted in Figs. 3–6, respectively. It can be found from these figures that all the values of Nu_b/Nu_0 and $fRe/(fRe)_0$ are greater than 1 which means an increase in both heat transfer and flow resistance induced by inserts. PEC > 1 indicates the heat transfer enhanced for given pumping power. It can also be observed that all the PEC curves have a similar trend as the corresponding Nu_b/Nu_0 curves.

The simulating values of Nu_b/Nu_0, $fRe/(fRe)_0$ and PEC for $\delta/D_e = 0.01$, 0.025, and 0.05 are plotted in Fig. 3. $fRe/(fRe)_0$ curves go up with the increase of the Reynolds number and show a weak dependence on δ/D_e in the range studied. This is because the tip of filament is located at the near-wall region where the flow velocity is rather slow, and so, variation of δ has only little

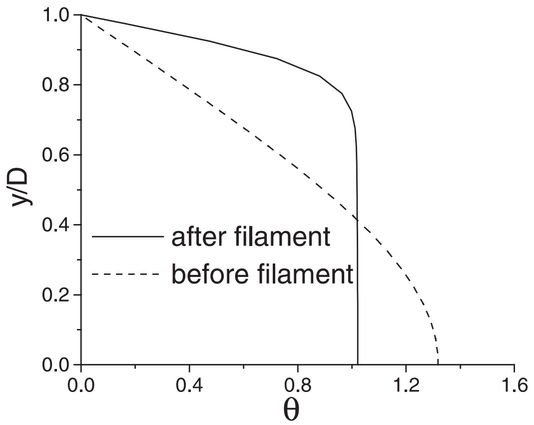

Fig. 2. Temperature profile of cross-section.

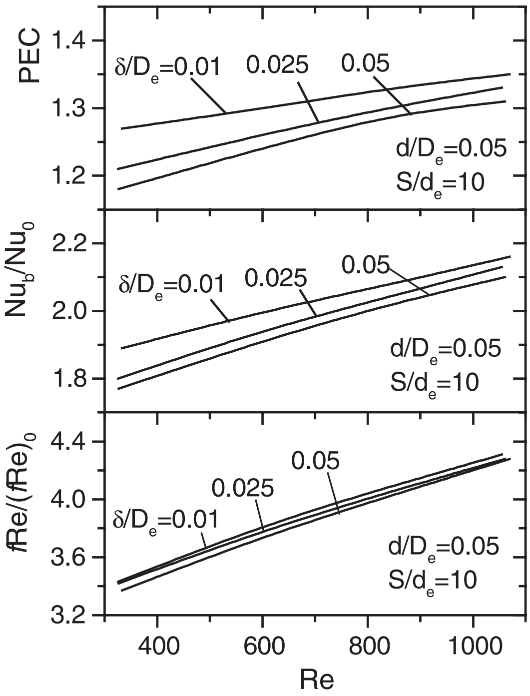

Fig. 3. Effect of δ on flow and heat transfer characteristics ($Pr = 0.7,\ \lambda_m = 400$).

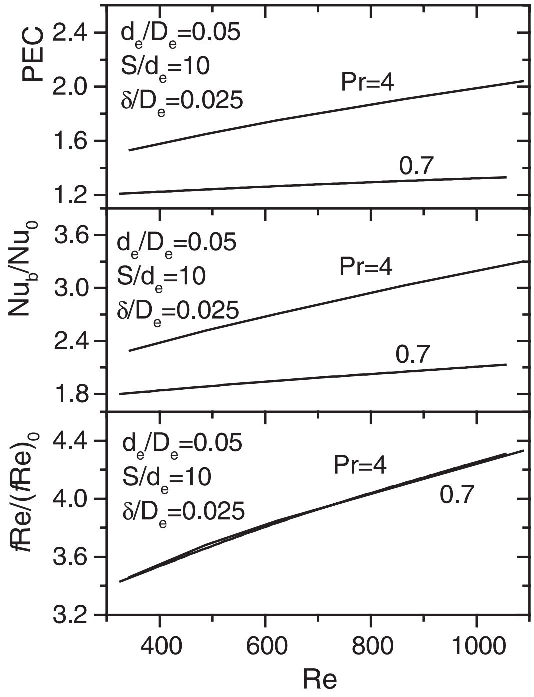

Fig. 5. Effect of Pr on flow and heat transfer characteristics ($\lambda_m = 400$).

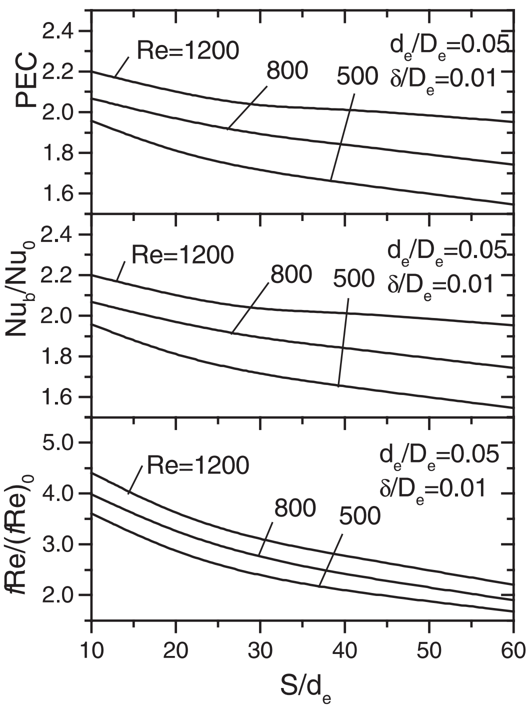

Fig. 4. Effect of S/d_e on flow and heat transfer characteristics ($Pr = 0.7,\ \lambda_m = 400$).

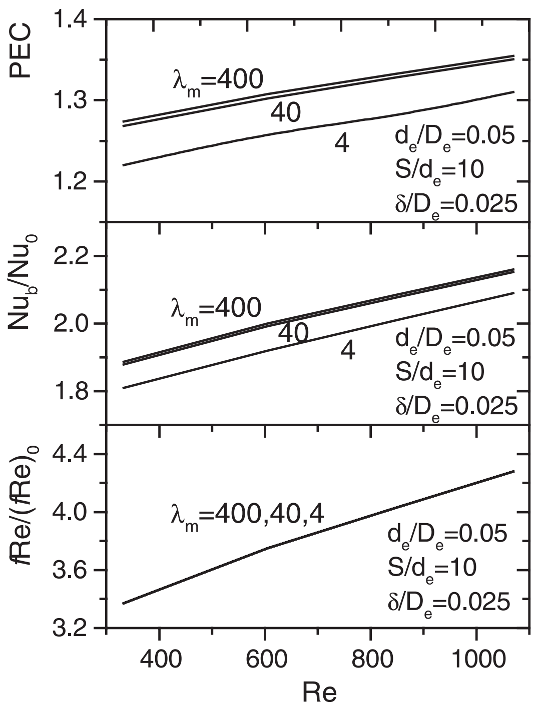

Fig. 6. Effect of λ_m on flow and heat transfer characteristics ($Pr = 0.7$).

influence on flow field. In addition, unlike smooth channel, *fRe* are dependent on Reynolds number here. Just the same, Nu_b/Nu_0 increases with increasing *Re*. However, compared with the neglectable effect of insert on flow characteristics, δ/D_e has a great impact on thermal characteristics, Nu_b/Nu_0 increases with decreasing δ/D_e. The reason lies in that the temperature profile of fluid flowing over the filament becomes more uniform as δ decreases and the temperature gradient gets more steep at the channel wall. So that, heat transfer is enhanced accordingly. As observed from Fig. 3, Nu_b/Nu_0 is dependent on δ/D_e and is stronger for lower Reynolds number. The same trends are seen for PEC curves.

Fig. 4 displays the variation of Nu_b/Nu_0, $fRe/(fRe)_0$ and PEC with S/d_e for three different values of Reynolds number. As S/d_e becomes very large, the behavior of filament insert channel approaches that of smooth channels, $fRe/(fRe)_0$ will approach to 1 as S/d_e increases, while Nu_b/Nu_0 tends to decrease with increasing S/d_e, In the range of S/d_e studied, Nu_b/Nu_0 decreases more rapidly for lower Reynolds number. A similar tendency can be observed for the PEC curves.

Fig. 5 shows the results of effect of fluid Prandtl number. Naturally, Prandtl number does not affect the flow resistance of filament insert channel. However, $Pr = 4$ fluid has a higher Nu_b/Nu_0 value than that of $Pr = 0.7$, so that, Nu_b/Nu_0 for $Pr = 4$ increases more quickly than that for $Pr = 0.7$, which shows a stronger dependence on Reynolds number for higher Prandtl numbers. Effect of λ_m is illustrated in Fig. 6. Clearly, λ_m does not alter flow field. But higher Nu_b/Nu_0 are observed for higher λ_m, and Nu_b/Nu_0 increases rapidly for a rather small λ_m ($\lambda_m = 4$) while these are approximately independent of λ_m for higher λ_m (for example, $\lambda_m = 40, 400$) at a given Reynolds number. The thermal conductivity of the metal is sufficiently high usually, so that metallic filament insert of various thermal conductivity has no remarkable influence on the heat transfer enhancement.

5. Experimental results

Experiments were conducted for air flowing through a square channel with copper filament insert. Since it is very difficult to fix the distance δ between the filament tip and wall surface in manufacture and assemblage, the contact between a few filaments and channel wall in the test channel is inevitable, and consequently, heat transfer will exist due to heat conduction through some contacted filaments. The hydraulic diameters of filament and channel are 0.5 and 10 mm, respectively. Pitch of the filaments is 2.5 mm and the length of channel is 200 mm. The sparse distribution of filaments makes up its volume porosity up to 99%. Schematic representation of the experimental rig and the associated instruments are displayed in Fig. 7. The test channel is placed horizontally in a boiling water bath, in which the test channel is about 60 mm over the boiling water surface. Channel wall is maintained at constant temperature and assumed to be 100°C. The test section is airproofed with only an outlet on the top to vent steam. The whole water bath is thermally insulated with blankets. Room air drawn by a fan passes through valves and the flow meter, then goes through the developing section, which is a channel 1.5 m long with the same cross-section as the test channel. Airflow is vented from test section into the surrounding environment. The inlet and outlet air temperatures and the wall temperature are measured with thermocouples,

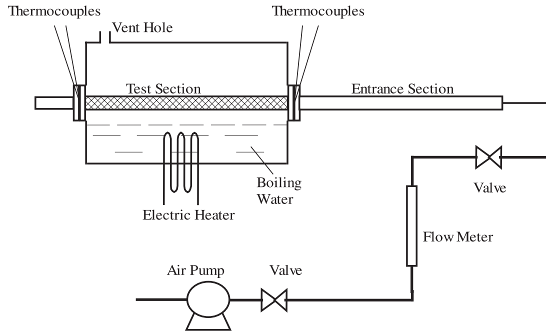

Fig. 7. Schematic representation of the experimental rig.

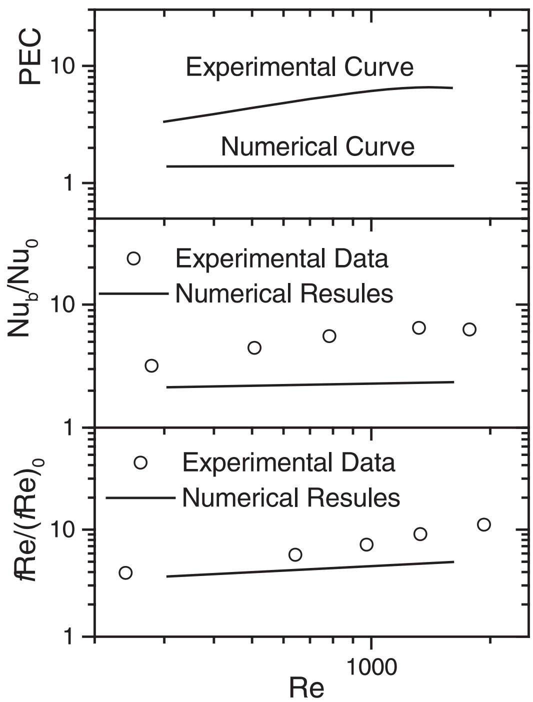

Fig. 8. Experimental and numerical results.

while flow resistance is measured with micro-pressure manometer.

The experimental results are plotted and compared with corresponding numerical results in Fig. 8. The imperfect contact between filament tips and channel wall is taken into account in numerical simulation for a specified $\delta = 0.01$ mm as defined in Fig. 1. Both experimental and numerical results show the similar tendency that, the higher the Reynolds number, the better the effect of heat transfer enhancement of inserts. Experimental heat transfer data are obviously higher than numerical results. Two factors should be responsible for this: contact of some filament with the wall and flow disturbance induced by filament. The latter one becomes stronger when *Re* increases. The numerical results for flow resistance agree well with the measured data, especially. For low *Re*, the experimental values of PEC, are far more than 1, and indicate that the filament insert channel can enhance heat transfer substantially for the given pumping power.

6. Concluding remarks

1. Based on the concept of heat transfer enhancement via improvement of the uniformity of temperature profile rather than the change of velocity field, a metallic filament insert has been developed, which is made of very large length-to-diameter ratio filaments normal to each other and set perpendicular to the flow direction and channel wall.
2. Numerical results indicate that the temperature profiles of fluid flowing over the filament become more uniform and the fluid temperature drops more sharply near the channel wall. As a result, heat transfer is enhanced.
3. The advantage of such an approach of heat transfer enhancement lies in the fact that heat transfer is augmented with less additional pressure drop. So, PEC can be as high as up to 3–8 for *Re* ranging from 200 to 1200.
4. Good agreement lies between numerical and experimental results of flow resistance. Two factors should be responsible for experimental heat transfer data higher than numerical results: One is the contact of some filaments with the channel wall, the other is the disturbance to the main flow by inserts, both of which are not to be taken into account in the numerical simulation.

References

[1] A.E. Bergles, Heat transfer enhancement – the encouragement and accommodation of high heat fluxes, ASME J. Heat Transfer 119 (1995) 8–19.
[2] A.E. Bergles, Application of Heat Transfer Augmentation, Hemisphere, Washington, DC, 1981.
[3] J.P. Gupta, Fundamentals of Heat Exchanger and Pressure Vessel Technology, Hemisphere, Washington, DC, 1985.
[4] R.L. Webb, Principles of Enhanced Heat Transfer, Hemisphere, Washington, DC, 1995.
[5] Z.Y. Guo, D.Y. Li, B.X. Wang, A novel concept for convective heat transfer enhancement, Int. J. Heat Mass Transfer 41 (1998) 2221–2225.
[6] S.V. Patankar, C.H. Liu, E.M. Sparrow, Fully developed flow and heat transfer in ducts: having streamwise-periodic variation of cross-sectional area, ASME J. Heat Transfer 99 (1977) 180–186.
[7] W.Z. Gu, J.R. Shen, C.F. Ma, Y.M. Zhang, Heat Transfer Enhancement, Scientific Press, Beijing, 1990.