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Enhancing convective heat transfer based on minimum power consumption principle

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ABSTRACT

Equation of power consumption in the flow process of micro-fluid elements has been deduced in this paper by analyzing the mechanical energy equilibrium equation of microelement based on the stress relation of Newtonian fluid. By setting minimum power consumption as optimization objective and entransy consumption as constraint condition, optimization flow field equation for convective heat transfer on the basis of energy conservation equation and continuity equation are constructed. Numerical investigations of forced convective heat transfer in a circular tube based on the optimized controlling equations are conducted, and the results show that there are multi-longitude-vortexes existing in velocity field, which leads to an enhancement of heat transfer. The analysis for the heat transfer and flow resistance shows that this kind of flow field accommodates a far greater heat transfer increase than flow resistance increase, which proves that the proposed optimization theory is effective in designing high efficient heat transfer enhancement unit.

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1. Introduction

In recent fifty years, a variety of enhanced heat transfer technology has been rapidly developed and widely applied, but many studies are based on experiments. The researchers design some kind of enhanced heat transfer component based on their experience, and then get a correlation, which is applied in certain range through experiment (Bergles, 1985, 1997). These correlations, which are obtained under certain circumstances, cannot serve as a universal guide of heat transfer enhancement; so it is very to important to study the theory of heat transfer enhancement.

With the development of heat transfer enhancement, the theory of heat transfer enhancement is also perfected step by step. Guo et al. (1998) proposed the field synergy principle in convective heat transfer while they studied the relation between local behavior and overall performance of 2-D laminar flow. He believed that the performance of convective heat tranfer was decided by the synergy of velocity field and the heat flux field. Under the same conditions of velocities and temperature boundaries, the better the synergy is, the higher the heat tranfer is. Xia (2001) set the thermal potential loss as the evaluation objective and used the viscous dissipation to denote the loss in mechanical energy. With fixed mechanical energy loss, the optimized velocity field equation can be derived through functional analysis, in which a scalar item was unknown. Meng et al. (2005) furthered Xia's analysis using Lagrange multiplier to construct functional analysis. The field synergy equation was thus derived, and the indication of each term in it was identified. Based on his analysis, two longitudinal vortex heat exchange enhancement tubes were designed, which exhibited ideal heat transfer performance. On the basis of field synergy heat transfer enhancement, Liu et al. (2009a, 2009b) studied the physical mechanism of laminar and turbulent convective heat transfer between the fluid and the wall and proposed the concept of physical quantity synergy in the laminar and turbulent flow field, which revealed relation between the law of field synergy of physical quantities and heat transfer enhancement, and explained the physical nature of the fluid of the heat transfer enhancement and flow resistance reduction; thus the flow resistance reduction will be placed in an important position in heat transfer enhancement. Bejan (1997, 1998) proposed and developed (Alebrahim and Bejan, 1999; Zimparov et al., 2006) the constructal theory, which refined the construction from large to small based on a fundamental unit. This approach simplified the complicated geometric optimization process into the assembly of a series of simple basic units, and enabled the optimization of specific conductive and flow problems. Bejan (1966, 1982) also deduced the entropy generation expression and analyzed the parameter optimization in heat exchanger and heat transfer system with the minimum entropy generation induced by viscous dissipation and heat transfer as optimization objective. Guo et al. (2007) proposed a new physical quantity—entransy to describe the heat transfer capability. He also proposed entransy dissipation

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extremum principles, which were minimum entransy dissipation principle in terms of temperature and maximum entransy dissipation principle in terms of heat flux (Cheng, 2004). Based on the entransy dissipation extremum principle, a lot of fruitful explorations by optimizing the whole heat transfer process with the fixed viscous dissipation as constraint conditions were made (Wu and Liang, 2008; Chen et al., 2007, 2008a, 2009, 2010).

Under the same power consumption, reducing the flow resistance in the process means that heat transfer process can be worked at the working conditions of higher velocity. The greater the fluid flow velocity, the stronger the heat transfer enhancement. The current heat transfer enhancement studies are mostly on how to enhance the heat transfer at certain inlet velocity. Many heat transfer enhancements have often led to a fact that an increase in the heat transfer coefficient often leads to an increase in the flow resistance, which increases heat losses than gains. Reducing the flow resistance of the process, not only means saving energy, more importantly, under the same energy consumption can often achieve higher heat transfer capacity, thus, reducing the flow resistance is also a heat transfer enhancement.

2. Power consumption of fluid microelement in flow process

Taking mechanical energy balance of a certain particle of homogeneous fluid into account, it can be found that the power done on the fluid within volume τ contains two aspects: the contribution of body force $\int u_i F_i \rho d\tau$ and the contribution of area force from the boundary $\int u_i \sigma_{ij} n_j dS = \int (\partial(u_i \sigma_{ij})/\partial x_j) d\tau$. Thus, for unit mass flow rate, power done on the certain fluid particle can be written as

$$u_i F_i + \frac{u_i}{\rho}\frac{\partial \sigma_{ij}}{\partial x_j} + \frac{\sigma_{ij}}{\rho}\frac{\partial u_i}{\partial x_j} = u_i \frac{Du_i}{Dt} + \frac{\sigma_{ij}}{\rho}\frac{\partial u_i}{\partial x_j} \tag{1}$$

Kinetic energy increase of fluid microelement $u_i Du_i/Dt$ consists of two items shown in the left hand of Eq. (1): the first term is power done by body force $u_i F_i$ and the second term relates to small difference of stress $(u_i/\rho)(\partial \sigma_{ij}/\partial x_j)$. The second term $(\sigma_{ij}/\rho)(\partial u_i/\partial x_j)$ on the right hand of Eq. (1) has a bearing on small difference of velocity, which appears as internal energy augment of fluid microelement induced by the deforming of fluid without changing its velocity (Batchelor, 1994). Substitute the stress relation of Newtonian fluid:

$$\sigma_{ij} = -p\delta_{ij} + 2\mu\left[e_{ij} - \tfrac{1}{3}(\nabla\cdot\mathbf{u})\delta_{ij}\right] \tag{2}$$

Then Eq. (1) can be expressed as

$$\begin{aligned}\frac{\sigma_{ij}}{\rho}\frac{\partial u_i}{\partial x_j} &= \frac{-p(\nabla\cdot\mathbf{u})}{\rho} + 2\mu\left[e_{ij}e_{ij} - \frac{1}{3}(\nabla\cdot\mathbf{u})^2\right] \\ &= \frac{1}{\rho}(-p\delta_{ij})\left[\frac{1}{3}(\nabla\cdot\mathbf{u})\delta_{ij}\right] + \frac{2\mu}{\rho}\left[e_{ij} - \frac{1}{3}(\nabla\cdot\mathbf{u})\right]\left[e_{ij} - \frac{1}{3}(\nabla\cdot\mathbf{u})\right]\end{aligned} \tag{3}$$

where the first term on the right hand of Eq. (3) is power done by isotropic part of stress or pressure related to isotropic part of strain rate or deformation of fluid; the second term is power done by deviation component of stress related to non-isotropic part of strain rate. The term $2\mu/\rho[e_{ij} - 1/3(\nabla\cdot\mathbf{u})][e_{ij} - 1/3(\nabla\cdot\mathbf{u})]$ is non-negative, showing that any shear movement of fluid inevitably can cause mechanical energy dissipating into internal energy unidirectional. So it is usually recorded as viscous dissipation Φ.

Therefore, power consumption of incompressible fluid flowing through pipe includes two sections: one is transformed to kinetic energy and the other is dissipated into thermal energy due to viscosity of fluid. Then it can be expressed as

$$P = \int_V S\left(\rho u_i \frac{Du_i}{Dt} + \Phi\right) dV \tag{4}$$

where Φ is

$$\begin{aligned}\Phi = \mu\Bigg\{ 2\left[\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial y}\right)^2 + \left(\frac{\partial w}{\partial z}\right)^2\right] + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)^2 + \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right)^2 \\ + \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y}\right)^2 \Bigg\}\end{aligned} \tag{5}$$

Considering the following item:

$$\begin{aligned}\mathbf{u}\cdot\mu\nabla^2\mathbf{u} &= \mu\Bigg(u\frac{\partial^2 u}{\partial x^2} + u\frac{\partial^2 u}{\partial y^2} + u\frac{\partial^2 u}{\partial z^2} + v\frac{\partial^2 v}{\partial x^2} + v\frac{\partial^2 v}{\partial y^2} \\ &\quad + v\frac{\partial^2 v}{\partial z^2} + w\frac{\partial^2 w}{\partial x^2} + w\frac{\partial^2 w}{\partial y^2} + w\frac{\partial^2 w}{\partial z^2}\Bigg) \\ &= \mu\nabla\cdot(u\nabla u + v\nabla v + w\nabla w) - \Phi\end{aligned} \tag{6}$$

It can be seen that when the velocity at boundary (namely wall) equals zero, Eq. (6) can be simplified into

$$\int_V \Phi\, dV = -\int_V u_i \cdot \mu\nabla^2 u_i\, dV \tag{7}$$

Thus, Eq. (4) can be expressed as

$$P = \int_V S\left(\rho u_i \frac{Du_i}{Dt} - u_i\cdot\mu\nabla^2 u_i\right) dV \tag{8}$$

From Eq. (8), it can be seen that the power consumption is solely correlated with velocity field; therefore, what we need to do is to find an optimal velocity field with a certain constraint, which satisfies the minimum power consumption.

3. Velocity field equation with the optimization objective of power consumption

To enhancement of convective heat transfer, not only we need to consider the enhancement of heat transfer, but also need to take the cost of intensity of heat transfer into account, namely power consumption in fluid flow process. A new physical quantity, which is termed as entransy consumption, can be used to describe the loss of heat transfer ability (Guo et al., 2007):

$$E_h = \lambda(\nabla T)^2 \tag{9}$$

The smaller the entransy consumption, the smaller the heat transfer ability wastage and the better the heat transfer process.

So when dealing with heat transfer enhancement problems, we consider such a question in present paper: how can we construct a flow field, which satisfies the power consumption minimum in fluid flow process and fulfills a certain heat exchange quantity. Based on the concept of power consumption and entransy consumption, therefore, optimizing the convective heat transfer process is to find an optimal velocity field with fixed entransy consumption constraint, which satisfies the minimum power consumption. This is a typical functional variational problem, and its detailed process is described below:

1) Optimization target: velocity field U

2) Optimization objective: power consumption extremum, expressed in terms of variation as

$$\delta\int_\Omega P\, dV = 0 \tag{10}$$

3) Constraint conditions:

Fixed entransy consumption:

$$\int_\Omega \lambda(\nabla T)^2 dV = const \tag{11}$$

Mass conservation:

$$\nabla \cdot U = 0 \tag{12}$$

Energy conservation:

$$\lambda \nabla^2 T - \rho C_p U \cdot \nabla T = 0 \tag{13}$$

4) Boundary conditions:

Constant velocity at boundaries, expressed in terms of variation as

$$\delta U|_\Gamma = 0 \tag{14}$$

Constant viscous shearing stress at boundaries (Yang, 1989), expressed in terms of variation as

$$\delta(\mu \nabla U)|_\Gamma = \delta(\nabla U)|_\Gamma = 0 \tag{15}$$

Constant wall temperature or heat flux boundaries, expressed in terms of variation as

$$\delta T|_\Gamma = 0 \text{ or } \delta(\lambda \nabla T)|_\Gamma = \delta(\nabla T)|_\Gamma = 0 \tag{16}$$

A function can be constructed by Lagrange multipliers as

$$J = \int_\Omega [U \cdot [\rho(U \cdot \nabla)U - \mu \nabla^2 U] + C_0 \lambda (\nabla T)^2 + A \nabla \cdot U + B(\lambda \nabla^2 T - \rho C_p U \cdot \nabla T)] dV \tag{17}$$

where C_0 is constant and A and B are unknown scalar functions. The variation of Eq. (17) yields (Qian, 1984)

$$\begin{aligned} \delta J = 0 = \int_\Omega \{ & (-2\lambda C_0 \nabla^2 T + \rho C_p U \cdot \nabla B + \lambda \nabla^2 B)\delta T \\ & + [\rho(U \cdot \nabla)U - \mu \nabla^2 U - \nabla A - \rho C_p B \nabla T]\delta U \\ & + (\nabla \cdot U)\delta A + (\lambda \nabla^2 T - \rho C_p U \cdot \nabla T)\delta B\} dV \\ & + \int_\Gamma \{[(2\lambda C_0 \nabla T - \rho C_p U B - \lambda \nabla B)\delta T + \lambda B \delta(\nabla T)] \\ & + [(\rho U \cdot U + \mu \nabla U + A)\delta U - \mu U \cdot \delta(\nabla U)]\} dS \end{aligned} \tag{18}$$

From Eq. (18), within the region Ω, we have

$$-2\lambda C_0 \nabla^2 T + \rho C_p U \cdot \nabla B + \lambda \nabla^2 B = 0 \tag{19}$$

$$\rho(U \cdot \nabla)U - \mu \nabla^2 U - \nabla A - \rho C_p B \nabla T = 0 \tag{20}$$

$$\nabla \cdot U = 0 \tag{21}$$

$$\lambda \nabla^2 T - \rho C_p U \cdot \nabla T = 0 \tag{22}$$

On the boundaries Γ, we have

$$(2\lambda C_0 \nabla T - \rho C_p U B - \lambda \nabla B)\delta T + \lambda B \delta(\nabla T) = 0 \tag{23}$$

$$(\rho U \cdot U + \mu \nabla U + A)\delta U - \mu U \cdot \delta(\nabla U) = 0 \tag{24}$$

With constraint conditions in Eqs. (12) and (13) and boundary conditions in Eqs. (14) and (15), it can be inferred that Eqs. (21), (22) and (24) hold true.

For momentum conservation, we have

$$\rho(U \cdot \nabla)U = F - \nabla p + \mu \nabla^2 U \tag{25}$$

By making an analogy (Chen et al., 2008b) between Eqs. (20) and (25), and let

$$A = -p \tag{26}$$

Then Eq. (20) can be converted into

$$\rho(U \cdot \nabla)U = -\nabla p + \mu \nabla^2 U + \rho C_p B \nabla T \tag{27}$$

where the unknown scalar B is determined by Eqs. (19) and (23). The last term on the right hand of Eq. (27) is an item of volume force adding to original momentum equation. It is the addition of this term that keeps the power consumption minimum when entransy consumption is fixed.

4. Analysis of the optimized field of laminar heat transfer in circular tubes

The enhancement of laminar heat transfer inside the circular tube, which is a common heat transfer unit used in tube-and-shell heat exchangers, will bring far-reaching benefits in engineering. In order to verify the effect of the optimization equation sets deduced above, a computational model is introduced, as shown in Fig. 1. The inside diameter D of the tube is 20 mm, and its total length is 1700 mm. The tube is divided into three sections: the entrance section L_1, the experimental section L_2 and the exit

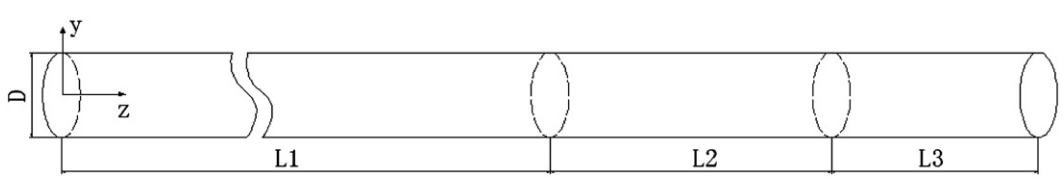

Fig. 1. Circular tube model.

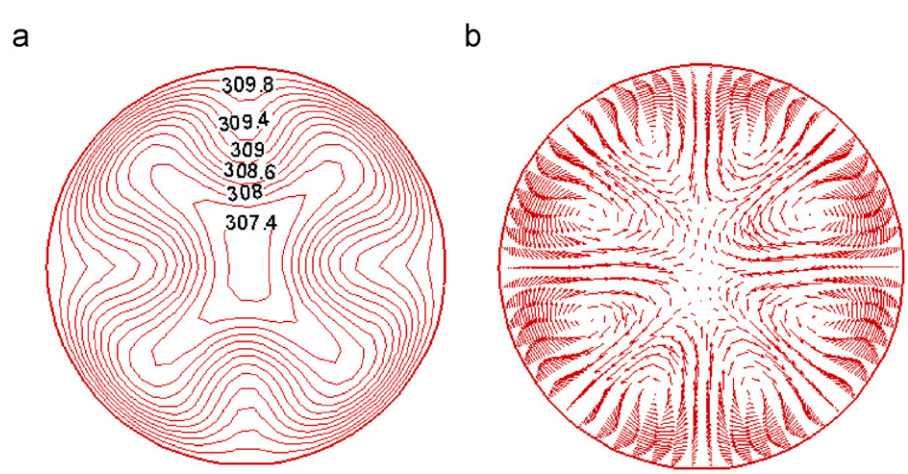

Fig. 2. Temperature field (a) and flow field (b) in the cross-section of tube (Re=200, C_0=1 × 10^{-9}).

section L_3, the lengths of which are 1200, 300, and 200 mm, respectively. The fluid flowing inside the tube is water of fixed physical quantity. The water temperature is 300 K and the wall of the entrance and experiment section is heated constantly and evenly at a temperature of 310 K.

The governing equations in numerical computation include continuity equation, energy conservation equation and velocity field optimization equation (see Eq. (27)). In the velocity field optimization equation, the governing equation and the boundary condition for the unknown scalar B is determined in Eqs. (19) and (23), respectively. The commercial software Fluent is used for solving the governing equations, and the SIMPLEC algorithm is used for coupling pressure field and velocity field. The momentum and energy equation applies QUICK discrete scheme. To solve the constraint scalars, we can use the user-defined functions supported by Fluent software.

We tried different values of entrance Reynolds numbers and C_0 for analysis purpose, and the changes occurring in the middle cross-section of the experiment section ($z=1350$ mm) were observed and analyzed. Fig. 2 is the cross-section chart of temperature field and flow field when $Re=200$, $C_0=1\times10^{-9}$, and Fig. 3 is the chart when $Re=200$, $C_0=6\times10^{-9}$. The observed results show that for the same entrance Re number, the solution varies with C_0 when it stays within a certain range. If beyond that range, for example, C_0 is too small, the source item would fail to function, or if too large (above a threshold value), it would be impossible to obtain convergence solutions. In the function, C_0 is the Lagrange coefficient corresponding to entransy consumption; so it can be used to represent the amount of entransy consumption. When C_0 is increasing within a specified range where convergence solutions could be obtained, Nu/Nu_s and f/f_s would increase monotonically with it, in which Nu_s and f_s are based on the model without additional volume force. This suggests that with the increase in C_0, the virtual volume force would increase and the flow field would assume a better organization, which would lead to better heat transfer performance. However, in the meantime, more power would be consumed, and the resistance coefficient would increase too. The trend of change is clearly demonstrated in Fig. 4, in which one can find that the optimized heat transfer enhancement effect occurs when $Re=200$ and $C_0=6\times10^{-9}$. At this point, the Nu number with additional volume force is 2.6 times as much as that without volume force, while f is only increased to 1.09 times, which suggests a far greater degree of heat transfer enhancement than resistance increase.

Fig. 5 is the cross-section chart of temperature field and flow field when $Re=500$, $C_0=5\times10^{-10}$, and Fig. 6 is the chart when $Re=500$, $C_0=2\times10^{-9}$. Fig. 7 displays the change in Nu number and resistance coefficient before and after the volume force is added when $Re=500$. From Fig. 7, one can find that the heat transfer enhancement degree is much greater than resistance increase degree when $Re=500$ and $C_0=2\times10^{-9}$. At this point, the

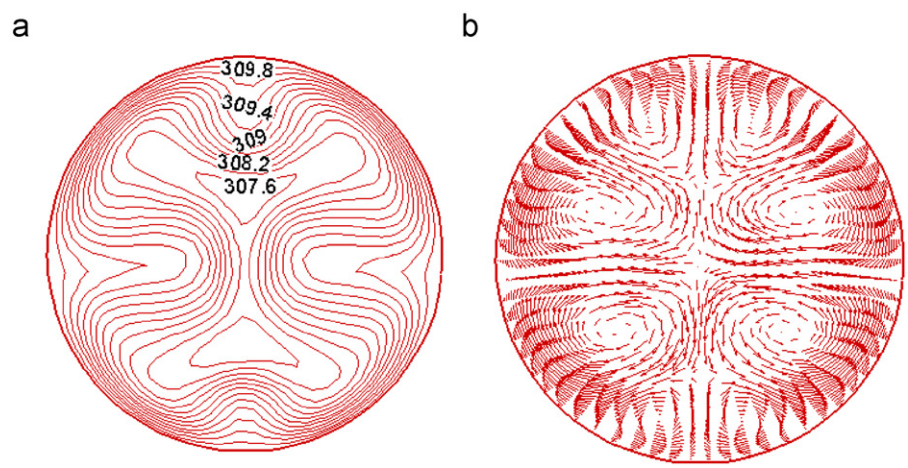

Fig. 3. Temperature field (a) and flow field (b) in the cross-section of tube ($Re=200$, $C_0=2.5\times10^{-9}$).

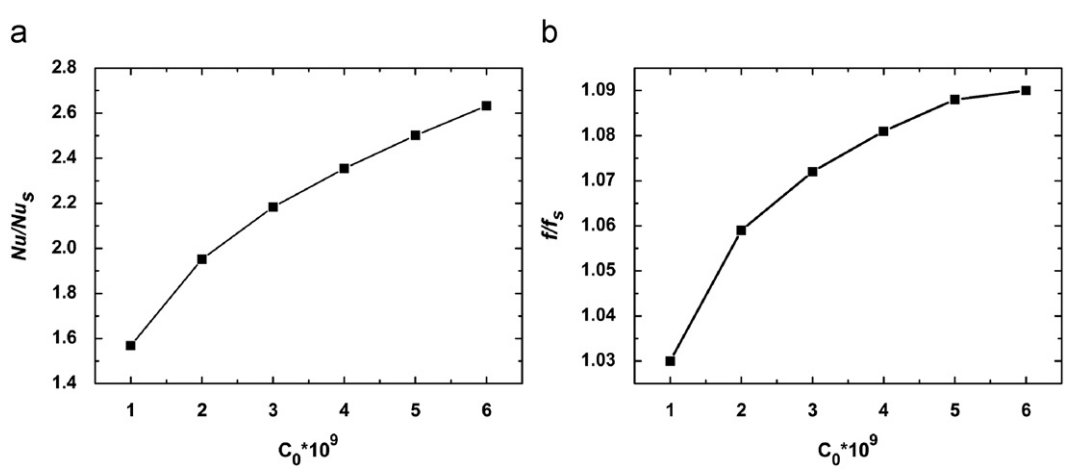

Fig. 4. Nu/Nu_s and f/f_s varying with C_0 ($Re=200$).

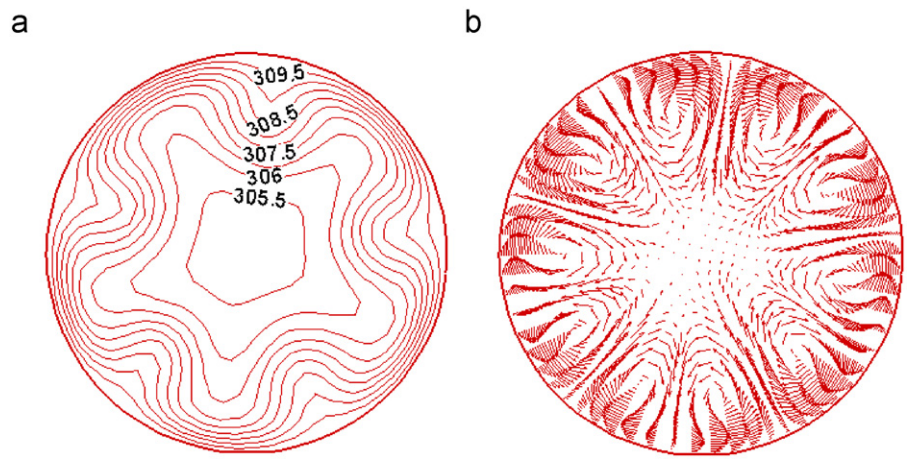

Fig. 5. Temperature field (a) and flow field (b) in the cross-section of tube ($Re=500$, $C_0=5\times10^{-10}$).

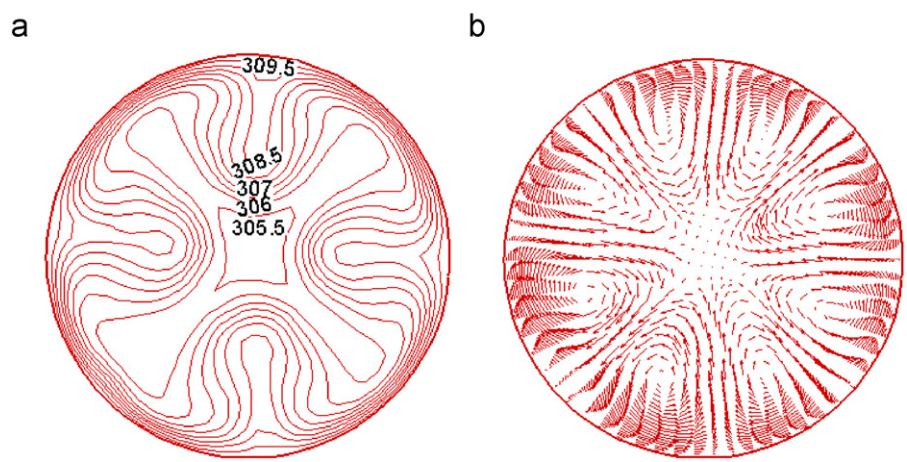

Fig. 6. Temperature field (a) and flow field (b) in the cross-section of tube ($Re=500$, $C_0=2\times10^{-9}$).

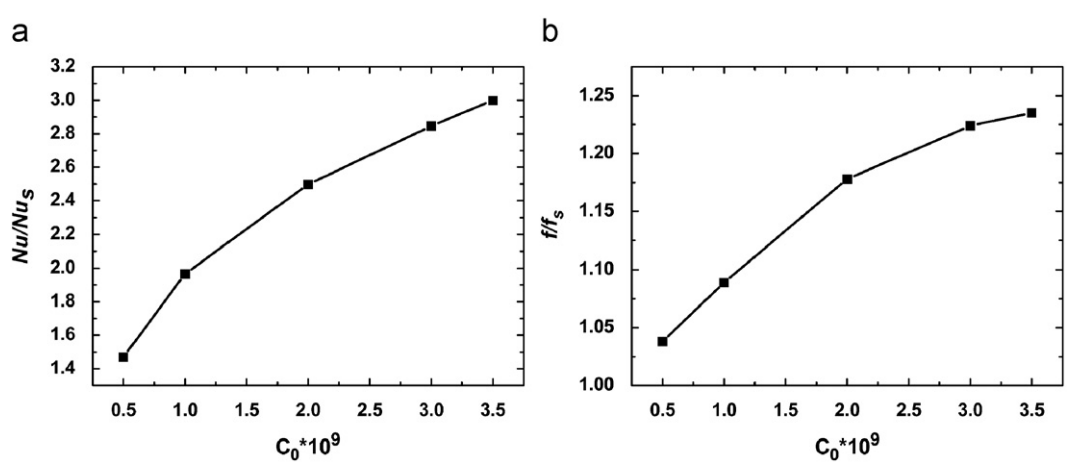

Fig. 7. Nu/Nu_s and f/f_s varying with C_0 ($Re=500$).

Nu number with additional volume force is 3 times as much as that without volume force, while the resistance coefficient is increased only to 1.25 times.

We also experimented with other *Re* numbers, and the results showed that for the circular tube with the size specified in this paper and at a fixed given *Re* number, when C_0 varied within a specified range where convergence solutions could be obtained, multi-vortexes in different shapes and structures would be formed in the optimized velocity field, and the number and shape of these vortexes would change with *Re* number. The multi-vortexes flow field would have far better heat transfer capability than a bare tube, with its heat transfer enhancement degree growing much faster than its flow resistance increase (Zhang et al., 2009; Lei et al., 2010).

The above analysis of the optimized temperature field reveals that the temperature gradient near tube wall is much greater than that in

the central tube. Apart from this, the analysis of the optimized velocity field demonstrates that vortexes are all distributed symmetrically around central area, leaving a comparatively small velocity gradient near the boundaries. A comparison between the increment of resistance coefficient f in Figs. 4 and 7 would make this point clear. This kind of flow field is consistent with the core flow enhancement principle proposed by Liu et al. (2007), Liu and Yang (2008), who intend to design a flow field in which the disturbance at the boundaries is kept at the lowest level while the disturbance and the temperature uniformity at the core area are enhanced to the greatest extent possible. Now when going back to review the optimization objective and the constraint conditions specified in the foregoing part of this paper, we find that the entransy consumption reflects the influence of thermal resistance, while the power consumption reflects that of the fluid flow resistance. So if we want to reconcile the heat transfer problem with the fluid flow problem, what we need to do is to decrease the thermal resistance as much as possible while keeping the flow resistance constant. In this way, the connotation of resistance deduction gains substantial embodiment.

5. Conclusion

Based on the stress relation of Newtonian fluid, consumption of power in the flow process of micro-fluid elements has been deduced in the present paper by analyzing the mechanical energy equilibrium equation of microelement.

By taking minimum power consumption as optimization objective and entransy consumption as fixed constraint condition, a generalized functional using Lagrange multiplier was constructed, from which the optimized velocity field equation was obtained by variational calculation.

Numerical solution of the flow field in the circular tube with an additional optimal volume force was analyzed. It is found that the multi-vortexes flow field in the core flow area present excellent performance in heat transfer enhancement with optimization objective and constraint conditions specified in this paper. *Nu* number can be increased by 20–200% compared to bare tube, while the flow resistance coefficient f is only increased by less than 25%. It is validated that the proposed optimization principle can provide strong theoretical guidance for designing high-efficiency and low-resistance heat transfer enhancement units.

Notation

A, B, C_0	Lagrange multipliers
c_p	specific heat at constant pressure (J/(kg K))
E_h	entransy consumption (W K/m^3)
F_i	component of volume force vector (N/m^3)
J	functional
P	power consumption (W)
p	pressure (Pa)
S	surface (m^2)
T	temperature (K)
t	time (s)
U, $\boldsymbol{u}$	velocity vector (m/s)
u_i, u, v, w	component of velocity vector (m/s)
V	volume (m^3)
x_j, x, y, z	component of coordinate (m)

Greek letters

λ	heat conductivity (W/(m K))
ρ	fluid density (kg/m^3)
μ	viscosity coefficient (kg/(m s))
Ω	control volume (m^3)
Γ	control surface (m^2)
Φ	viscous dissipation (W/m^3)
σ_{ij}	component of stress tensor
δ_{ij}	component of normal stress tensor
e_{ij}	component of deviatoric stress tensor

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