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# Physical quantity synergy in laminar flow field of convective heat transfer and analysis of heat transfer enhancement

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**Based on the principle of field synergy for heat transfer enhancement, the concept of physical quantity synergy in the laminar flow field is proposed in the present study according to the physical mechanism of convective heat transfer between fluid and tube wall. The synergy regulation among physical quantities of fluid particle is revealed by establishing formulas reflecting the relation between synergy angles and heat transfer enhancement. The physical nature of enhancing heat transfer and reducing flow resistance, which is directly associated with synergy angles $\alpha$, $\beta$, $\gamma$, $\phi$, $\theta$ and $\psi$, is also explained. Besides, the principle of synergy among physical quantities is numerically verified by the calculation of heat transfer and flow in a thin cylinder-interpolated tube, which may guide the optimum design for better heat transfer unit and high-efficiency heat exchanger.**

physical quantity synergy, heat transfer enhancement, laminar flow, convective heat transfer, numerical verification



Heat exchangers are widely used in various industrial sectors such as power generation, chemical engineering, petroleum refining, steel and iron, metallurgy, refrigeration, air-conditioning, cryogenic engineering, and so on. Many researchers have been working hard on heat transfer enhancement to improve the performance of heat exchangers in the aspects of enhancing convective heat transfer and reducing flow resistance. The most commonly used methods for heat transfer enhancement at present are reducing boundary layer thickness, increasing heat transfer area and strengthening fluid disturbance near the solid wall[1,2]. The corresponding techniques for heat transfer enhancement include helical corrugation tube, transverse corrugation tube, longitudinal corrugation tube, enhanced tube with longitudinal vortex generators, inner-finned tube, finned tube with lower thread, and so on. Besides, the tube inserts, such as twisted tape, wire matrix, wire coil and metal foam, are also popular ways for heat transfer enhancement in a tube.

During the past decades, techniques for heat transfer enhancement have been developed rapidly, while related theories have also been improved. Based on energy conservation equation, Guo et al.[3] afresh surveyed the physical mechanism of convective heat transfer and developed the field synergy principle for enhancing heat transfer. They proposed a concept that physical nature of convective heat transfer is up to the synergetic relation between its velocity field and heat-flux field. Under the same boundary conditions of velocity and temperature, the better the synergy between velocity field and heat-flux field is, the higher the heat transfer intensity will be. Since the field synergy principle can demonstrate how the synergetic relation between fluid velocity and heat flux affects heat transfer in the flow field, a unified cri-

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terion for designing heat transfer unit towards enhancing convective heat transfer was established. Ref. [4] presented a concept of synergy between velocity and velocity gradient, and discussed the effect of synergy on reducing flow resistance. Some numerical computations and experiment data in refs. [5—22] verified that the field synergy principle can be used as a guide to design heat transfer surfaces and heat exchangers.

In the laminar flow field of single-phase convective heat transfer, there exist basic physical quantities such as temperature, velocity and pressure, which are continuously differentiable, and their corresponding derivatives such as temperature gradient, velocity gradient and pressure gradient. Furthermore, the magnitude and direction of those physical quantities in the laminar flow field will determine intensity of convective heat transfer and power consumption of a heat exchanger. So it is necessary to seek for the synergetic regulation among more physical quantities, which is closely related to heat transfer enhancement.

## 1 Principle of physical quantity synergy in nonisothermal flow field

For the problem of forced convective heat transfer between fluid and solid surface in tube flow or channel flow, a main flow should be maintained in order to effectively remove heat from the solid surface (or the fluid). As for velocity vector of fluid, its component in the longitudinal direction dominates convective heat transfer, while its components in other directions are relatively smaller. For a constant stream speed, if fluid flow is strengthened in the longitudinal direction, flow resistance of fluid will decrease.

### 1.1 Synergy between velocity gradient and temperature gradient

For steady laminar heat transfer in a two-dimensional parallel channel with height $H$ and length $L$, if it is analyzed symmetrically by taking $h=H/2$ channel length, then its energy conservation equation can be described as

$$\rho c_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)=\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right). \tag{1}$$

Integrating eq. (1) along $y$ direction within the boundary layer yields

$$\int_0^{\delta_t}\rho c_p\left(u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}\right)\mathrm{d}y=\int_0^{\delta_t}\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)\mathrm{d}y=-k\left.\frac{\partial T}{\partial y}\right|_w . \tag{2}$$

The non-dimensional numbers are defined as

$$Y=\frac{y}{h},\ \ \bar{\boldsymbol{U}}=\frac{\boldsymbol{U}}{u_m},\ \ \nabla\bar{T}=\frac{\nabla T}{(T_w-T_m)/h},\ \ T_w>T_m ,$$

where $h$ refers to half channel height, $\boldsymbol{U}$ refers to fluid velocity vector, $u_m$ refers to fluid average velocity, $T_w$ refers to channel wall temperature, $T_m$ refers to fluid average temperature.

Thus, eq. (2) can be expressed as the following non-dimensional form[3]:

$$Nu=RePr\int_0^{\delta_t/h}\left(\bar{\boldsymbol{U}}\cdot\nabla\bar{T}\right)\mathrm{d}Y , \tag{3}$$

where Reynolds number is $Re=u_m h/\nu$, Prandlt number is $Pr=\rho c_p\nu/k$. In fact, if the boundary layer merges in the center plane of channel, integral limit becomes $\delta_t/h=1$, which means the fluid enters the fully developed region. So eq. (3) can be applied in the whole channel.

In eq. (3), dot product of non-dimensional velocity and non-dimensional temperature gradient can be expressed as[3]

$$\bar{\boldsymbol{U}}\cdot\nabla\bar{T}=\left|\bar{\boldsymbol{U}}\right|\left|\nabla\bar{T}\right|\cos\beta . \tag{4}$$

After substituting eq. (4) into eq. (3), we can know that dot product $\bar{\boldsymbol{U}}\cdot\nabla\bar{T}$ increases with the decrease of synergy angle $\beta$ between vectors $\boldsymbol{U}$ and $\nabla T$. Obviously, the increase of dot product $\bar{\boldsymbol{U}}\cdot\nabla\bar{T}$ means the increase of $Nu$ number, and as a result, convective heat transfer between fluid and solid wall will be enhanced. In other words, if the direction of fluid velocity is closer to that of heat flux, the effect of convective heat transfer will be better in the laminar flow field.

### 1.2 Synergy between velocity and velocity gradient

For the laminar flow in the two-dimensional parallel channel mentioned above, its momentum conservation equation is

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial p}{\partial x}+\frac{\partial}{\partial y}\left(\mu\frac{\partial u}{\partial y}\right). \tag{5}$$

Integrating eq. (5) along the $y$ direction within the boundary layer yields

$$\int_0^{\delta}\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)\mathrm{d}y=-\int_0^{\delta}\frac{\partial p}{\partial x}\mathrm{d}y+\int_0^{\delta}\frac{\partial}{\partial y}\left(\mu\frac{\partial u}{\partial y}\right)\mathrm{d}y$$

$$= -\int_0^{\delta} \frac{\partial p}{\partial x} \mathrm{d}y - \mu \frac{\partial u}{\partial y}\bigg|_w . \quad (6)$$

By integrating eq. (6) along the $x$ direction from channel inlet to its outlet, we obtain

$$\int_0^L \int_0^{\delta} \rho(\boldsymbol{U} \cdot \nabla u)\mathrm{d}x\mathrm{d}y = -\int_0^L \int_0^{\delta} \frac{\partial p}{\partial x}\mathrm{d}x\mathrm{d}y - \int_0^L \tau_w \mathrm{d}x, \quad (7)$$

where $\tau_w$ stands for shear stress of the wall, and integral of viscous force on channel wall is

$$\int_0^L \tau_w \mathrm{d}x = \int_0^{L_1} \tau_{w_1} \mathrm{d}x + \int_{L_1}^L \tau_{w_2} \mathrm{d}x, \quad (8)$$

where $\tau_{w_1}$ and $\tau_{w_2}$ represent the shear stresses in the channel entrance region and the fully developed flow region respectively. According to Ref. [23], we can obtain $\tau_{w_1}$ and $\tau_{w_2}$ as:

$$\tau_{w_1} = \frac{0.323\rho u_m^2}{\sqrt{Re}\sqrt{x/h}}, \quad x < L_1, \quad (9)$$

$$\tau_{w_2} = \frac{3\rho u_m^2}{Re}, \quad x \geqslant L_1, \quad (10)$$

where $L_1$ stands for length of the channel entrance region.

Substituting eqs. (9), (10) into eq. (8) first, and then substituting eq. (8) into eq. (7), we have

$$\int_0^L \int_0^{\delta} \rho(\boldsymbol{U} \cdot \nabla u)\mathrm{d}x\mathrm{d}y = -\int_0^L \int_0^{\delta} \frac{\partial p}{\partial x}\mathrm{d}x\mathrm{d}y - \frac{0.646\rho u_m^2 L_1}{\sqrt{Re}\sqrt{L_1/h}} - \frac{3\rho u_m^2 (L - L_1)}{Re}. \quad (11)$$

The non-dimensional numbers are introduced as

$$X = \frac{x}{L}, \; Y = \frac{y}{h}, \; \bar{\boldsymbol{U}} = \frac{\boldsymbol{U}}{u_m}, \; \bar{u} = \frac{u}{u_m},$$

$$Eu = \Delta\bar{p} = \frac{\Delta p}{\rho u_m^2}, \; \chi_1 = \frac{L_1}{L}, \; \chi_2 = \frac{L - L_1}{L},$$

$$\nabla\bar{u} = \frac{\left(\frac{\partial}{\partial x}\boldsymbol{i} + \frac{\partial}{\partial y}\boldsymbol{j}\right)u}{u_m/h}, \; \nabla\bar{p} = \frac{\left(\frac{\partial}{\partial x}\boldsymbol{i} + \frac{\partial}{\partial y}\boldsymbol{j}\right)p}{\rho u_m^2/h},$$

$$\frac{\partial}{\partial y}\boldsymbol{j} = 0,$$

where $Eu$ refers to Euler number, $\Delta p$ refers to pressure drop between channel inlet and outlet, $\boldsymbol{i}$ and $\boldsymbol{j}$ refer to unit vectors of $x$ and $y$ coordinates respectively.

Then equation (11) can be expressed as

$$\int_0^1 \int_0^{\delta/h} (\bar{\boldsymbol{U}} \cdot \nabla\bar{u})\mathrm{d}X\mathrm{d}Y = -\int_0^1 \int_0^{\delta/h} (\nabla\bar{p} \cdot \boldsymbol{I})\mathrm{d}X\mathrm{d}Y - \frac{0.646\chi_1}{\sqrt{Re}\sqrt{L_1/h}} - \frac{3\chi_2}{Re}, \quad (12)$$

where $\delta/h$ refers to non-dimensional thickness of velocity boundary layer. If the velocity boundary layer merges in the center plane of channel, integral limit becomes $\delta/h = 1$, which means the channel flow becomes fully developed. $\boldsymbol{I}$ refers to unit vector. Integral term on the right-hand side refers to non-dimensional pressure drop for the parallel channel with height $h=H/2$, which can be expressed as

$$\Delta\bar{p} = -\int_0^1 \int_0^{\delta/h} (\nabla\bar{p} \cdot \boldsymbol{I})\mathrm{d}X\mathrm{d}Y. \quad (13)$$

From eqs. (12) and (13), an expression for $Eu$ number can be deduced as

$$Eu = \frac{0.646\chi_1}{\sqrt{Re}\sqrt{L_1/h}} + \frac{3\chi_2}{Re} + \int_0^1 \int_0^{\delta/h} (\bar{\boldsymbol{U}} \cdot \nabla\bar{u})\mathrm{d}X\mathrm{d}Y, \quad (14)$$

where dot product of non-dimensional velocity and non-dimensional velocity gradient can be expressed as

$$\bar{\boldsymbol{U}} \cdot \nabla\bar{u} = |\bar{\boldsymbol{U}}||\nabla\bar{u}|\cos\alpha. \quad (15)$$

After substituting eq. (15) into eq. (14), we can find that dot product $\bar{\boldsymbol{U}} \cdot \nabla\bar{u}$ decreases with the increase of synergy angle $\alpha$ between vectors $\boldsymbol{U}$ and $\nabla u$. But it can be noted that the decrease of dot product $\bar{\boldsymbol{U}} \cdot \nabla\bar{u}$ leads to the decrease of $Eu$ number, and as a result, the flow resistance of fluid will decrease.

For momentum conservation equation of the $y$ direction in parallel channel, similar expression can be obtained as

$$\Delta p_y = \int_0^L \int_0^{\delta} \rho(\boldsymbol{U} \cdot \nabla v)\mathrm{d}x\mathrm{d}y - \int_0^L \int_0^{\delta} (\mu\nabla^2 v)\mathrm{d}x\mathrm{d}y, \quad (16)$$

where the second term in the right-hand side is fluid viscous dissipation. And the total pressure loss of the $y$ direction in the channel is

$$\Delta p_y = -\int_0^L \int_0^{\delta} \frac{\partial p}{\partial y}\mathrm{d}x\mathrm{d}y. \quad (17)$$

The dot product of the first term in the right-hand side of eq. (16) can be expressed as

$$\boldsymbol{U} \cdot \nabla v = |\boldsymbol{U}||\nabla v|\cos\psi. \quad (18)$$

After substituting eq. (18) into eq. (16), we can find that dot product $\boldsymbol{U} \cdot \nabla \boldsymbol{v}$ decreases with the increases of synergy angle $\psi$ between vectors $\boldsymbol{U}$ and $\nabla\boldsymbol{v}$. The decrease of dot product $\boldsymbol{U} \cdot \nabla\boldsymbol{v}$ means the decrease of flow resistance in the direction of non mainstream, and as a result, the total flow resistance of fluid in the $y$ direction of parallel channel will decrease.

### 1.3 Synergy among velocity, temperature gradient and velocity gradient

A nonisothermal flow field of convective heat transfer is aggregated by numberless fluid particles, and each fluid particle has different physical quantities including scalar and vector. As we know that scalars, such as temperature *T* and pressure *P*, have definite physical meanings but no directions, so there is no direct synergy relation among them. However, scalar gradient and velocity vector of a fluid particle in the flow field reflect not only intensity of heat transfer, but also direction of transport process, so the coupling of those vectors represents the direct synergy relation. Therefore, revealing the physical mechanism of physical quantity synergy can help to explain heat transfer and flow process.

From eqs. (4) and (15), the synergy angles among velocity, velocity gradient and temperature gradient of a fluid particle *M* in the laminar flow field can be written as

$$\alpha = \arccos\frac{\boldsymbol{U}\cdot\nabla u}{|\boldsymbol{U}||\nabla u|}, \tag{19}$$

$$\beta = \arccos\frac{\boldsymbol{U}\cdot\nabla T}{|\boldsymbol{U}||\nabla T|}. \tag{20}$$

According to vector relation of a fluid particle *M*, the synergy angle between temperature gradient $\nabla T$ and velocity gradient $\nabla u$ can be expressed as

$$\gamma = \arccos\frac{\nabla T\cdot\nabla u}{|\nabla T||\nabla u|}. \tag{21}$$

For the two-dimensional laminar flow field, vectors $\boldsymbol{U}$, $\nabla T$ and $\nabla u$ are coplanar, thus all fluid particles on the stream line satisfy with $\gamma \equiv |\alpha-\beta|$. For the three-dimension laminar flow field, vectors $\boldsymbol{U}$, $\nabla T$ and $\nabla u$ are noncoplanar, and this will result in $\gamma \neq |\alpha-\beta|$.

The pressure gradient $\nabla p$ not only drives the flow of fluid, but also affects heat transfer and flow process. So, it can be inferred that, for a fluid particle, there exists the synergy relation between pressure gradient $\nabla p$ and velocity gradient $\nabla u$, which can be expressed as

$$\phi = \arccos\frac{\nabla p\cdot\nabla u}{|\nabla p||\nabla u|}. \tag{22}$$

For potential flow with no viscosity in a parallel channel, $\nabla p$ is orthogonal with $\nabla u$, $\phi$ = 90°, so flow resistance is zero. For viscous flow, there must be $\phi$<90° due to viscous dissipation. Moreover, viscous dissipation will increase with the decrease of synergy angle $\phi$, so flow resistance will increase.

In the same way, synergy angle between velocity $\boldsymbol{U}$ and pressure gradient $\nabla p$ can be expressed as

$$\theta = \arccos\frac{U\cdot\nabla p}{|U||\nabla p|}. \tag{23}$$

For the two-dimensional laminar flow field, vectors $\boldsymbol{U}$, $\nabla p$ and $\nabla u$ are coplanar, thus all fluid particles on the stream line satisfy $\phi \equiv |\alpha-\theta|$. For the three-dimension laminar flow field, vectors $\boldsymbol{U}$, $\nabla p$ and $\nabla u$ are noncoplanar, and this will result in $\phi \neq |\alpha-\theta|$.

From eq. (23), it can be seen that the smaller the synergy angle $\theta$ is, the better the synergy between $\boldsymbol{U}$ and $\nabla p$ will be. This will result in the decrease of flow resistance. It is worthy to point out that for the optimum design of a heat exchanger, regulating synergy angle $\theta$ for reducing flow resistance plays the same role as improving synergy angle $\beta$ for enhancing heat transfer. The better the synergy between velocity $\boldsymbol{U}$ and driving potential $\nabla p$ is, the smaller the pressure drop will be. This means fluid power dissipation in a heat exchanger will be smaller.

So far, the fully synergy correlation among velocity $\boldsymbol{U}$, velocity gradient $\nabla u$, temperature gradient $\nabla T$ and pressure gradient $\nabla p$ is obtained, in which $\nabla u$ serves as a reference vector, and it is schematically shown in Figure 1. For the situation of engineering application, it is a common way to fully mix the fluid for getting more uniform temperature. Thus the direction of velocity gradient $\nabla v$ will change, and the relation among $\nabla v$ and other vectors will be complex. The present study mainly

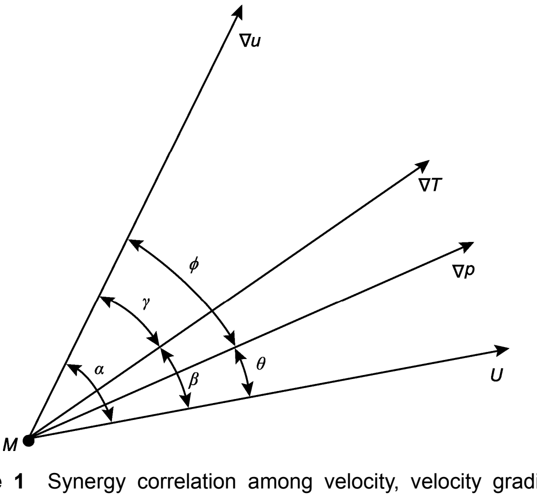


**Figure 1** Synergy correlation among velocity, velocity gradient, temperature gradient and pressure gradient for fluid particle *M*.

focuses on the synergy relation among velocity gradient $\nabla u$ and other vectors to reveal the synergy regulation among physical quantities for the entire flow field.

It can be observed from Figure 1 that for a certain fluid particle $M$, there are five intersection angles to reflect the synergy relation among physical quantities. It can be predicted that, if the synergy relation of every fluid particle in the entire fluid field is improved, then heat transfer and flow can be effectively organized to achieve better temperature uniformity in the flow field, and flow resistance can also be reduced. Actually, there are three kinds of problems in terms of heat transfer enhancement. (i) If the convective heat transfer is to be enhanced, then the synergy between velocity $\boldsymbol{U}$ and temperature gradient $\nabla T$ should be considered. The smaller the synergy angle $\beta$ is, the bigger the convective heat transfer coefficient $h$ will be. (ii) If the flow resistance is to be reduced, then the synergy between velocity $\boldsymbol{U}$ and pressure gradient $\nabla p$ should be considered. The smaller the synergy angle $\theta$ is, the smaller the pressure drop $\Delta p$ will be. (iii) If the purpose is to raise the overall performance of a heat transfer unit, then the synergy between temperature gradient $\nabla T$ and velocity gradient $\nabla u$ should be considered. The bigger the synergy angle $\gamma$ is, the higher the PEC value will be. PEC is a general evaluation coefficient that indicates the comprehensive performance of a heat transfer unit, which is commonly defined as

$$\mathrm{PEC} = \frac{Nu / Nu_0}{(f / f_0)^{1/3}}, \tag{24}$$

where $Nu_0$ and $f_0$ stand for Nusselt number and fluid resistance coefficient in parallel channel or bare tube respectively.

## 2 Numerical verification for principle of physical quantity synergy in noniso-thermal flow field

On the basis of analysis method for field synergy[3], we have established the expression of synergy among physical quantities represented by synergy angles $\alpha$, $\beta$, $\gamma$, $\phi$, $\theta$ and $\psi$, and analyzed their corresponding meaning on heat transfer enhancement. Although the deduction is aimed at the laminar flow in the two-dimensional parallel channel, the expression is representative and can be applied to other two- and three-dimensional problems.

### 2.1 Physical and mathematical model

Based on the physical model in which thin cylinders for disturbing fluid are interpolated in a tube, numerical verification for the synergy principle of physical quantities is carried out. As shown in Figure 2, we consider a three-dimensional tube with diameter $D$ = 20 mm and length $L$ = 500 mm respectively. The thin cylinders with diameter $d$ = 1 mm and length $l$ = 18 mm are distributed sparsely with pitch $s$ = 25 mm in the tube, which are perpendicular to each other, and vertical to the flow direction and wall surface.

The general control equation for the above model is

$$\frac{\partial(\rho u \Phi)}{\partial x} + \frac{\partial(\rho v \Phi)}{\partial y} + \frac{\partial(\rho w \Phi)}{\partial z} = \frac{\partial}{\partial x}\left(\Gamma \frac{\partial \Phi}{\partial x}\right) + \frac{\partial}{\partial y}\left(\Gamma \frac{\partial \Phi}{\partial y}\right) + \frac{\partial}{\partial z}\left(\Gamma \frac{\partial \Phi}{\partial z}\right) + S, \tag{25}$$

where $\rho$ refers to fluid density, $u$, $v$ and $w$ refer to fluid velocity components along $x$, $y$ and $z$ directions respectively, $\Gamma$ refers to generalized diffusion coefficient defined in ref. [24], $S$ refers to source term with different meanings in the different equations, $\Phi$ refers to the generalized variable to be displaced by $\Phi$ = 1 for continuity equation, $\Phi$ = $u$, $v$ and $w$ for momentum equation, and $\Phi$ = $T$ for energy equation.

### 2.2 Results and discussions

In this paper, the finite difference method and the two-order upwind difference scheme are applied to the numerical computation, and the SIMPLE algorithm is used for the coupling between pressure and velocity. The boundary values for numerical simulation are set as: wall temperature of parallel channel $T_w$ = 350 K, and inlet temperature of fluid $T_\infty$ = 293 K. The computa-

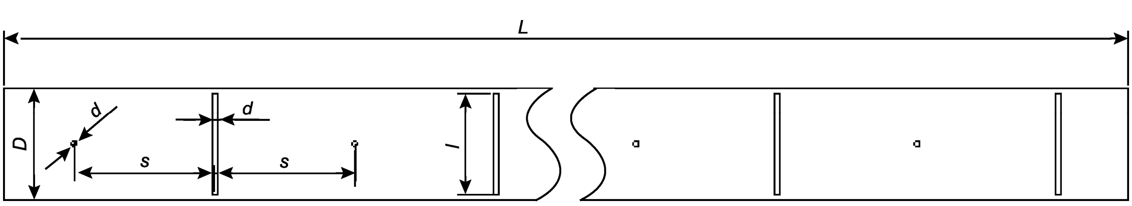


**Figure 2** Physical model of thin cylinder-interpolated tube.

tional fluid is water and its physical properties are kept as constant. The computational results are shown in Figures 3－11.

Figure 3 shows the effect of *Re* number on average synergy angle $\alpha$ for bare tube and thin cylinder-interpolated tube. As shown in the figure, average synergy angle $\alpha$ between fluid velocity $\boldsymbol{U}$ and velocity gradient $\nabla u$ in thin cylinder-interpolated tube is smaller than that in bare tube, so it can be known from eq. (14) that flow resistance of fluid will increase. Figure 4 shows the effect of *Re* number on average synergy angle $\beta$ for bare

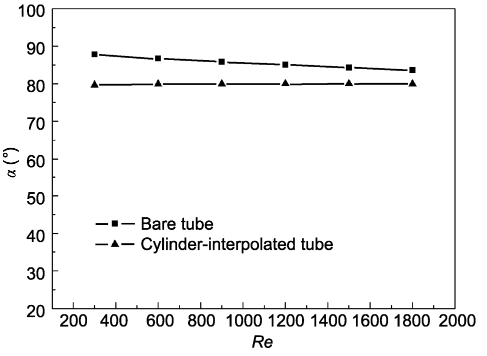


**Figure 3** Relation between *Re* number and average synergy angle $\alpha$ in bare tube and thin cylinder-interpolated tube.

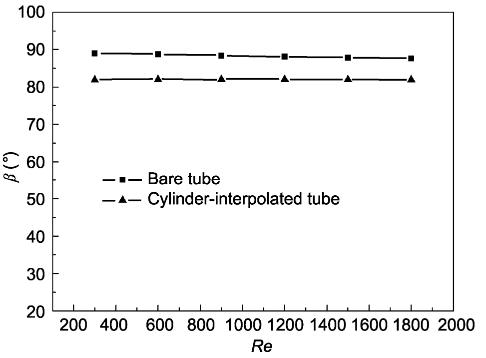


**Figure 4** Relation between *Re* number and average synergy angle $\beta$ in bare tube and thin cylinder-interpolated tube.

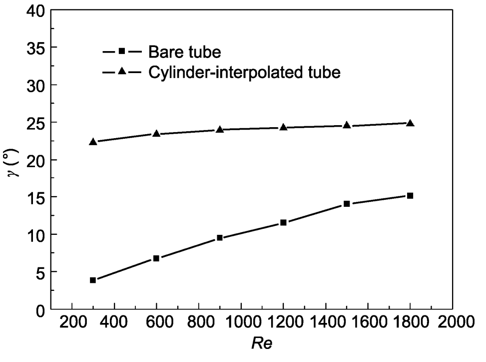


**Figure 5** Relation between *Re* number and average synergy angle $\gamma$ in bare tube and thin cylinder-interpolated tube.

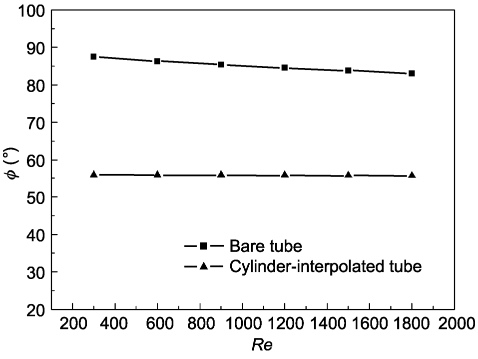


**Figure 6** Relation between *Re* number and average synergy angle $\phi$ in bare tube and thin cylinder-interpolated tube.

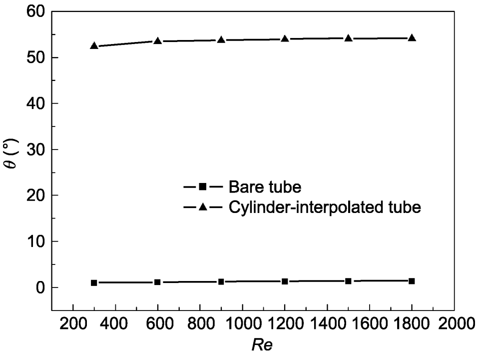


**Figure 7** Relation between *Re* number and average synergy angle $\theta$ in bare tube and thin cylinder-interpolated tube.

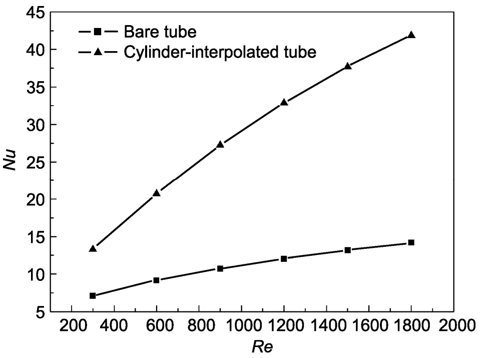


**Figure 8** Relation between *Re* number and *Nu* number of fluid in bare tube and thin cylinder-interpolated tube.

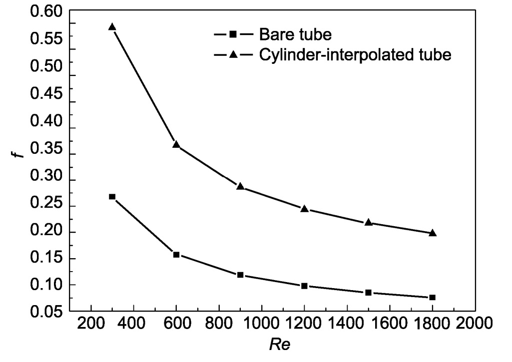


**Figure 9** Relation between *Re* number and resistance coefficient *f* of fluid in bare tube and thin cylinder-interpolated tube.

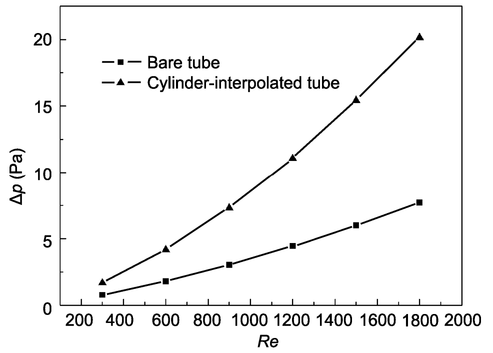


**Figure 10** Relation between *Re* number and pressure drop ∇*p* of fluid in bare tube and thin cylinder-interpolated tube.

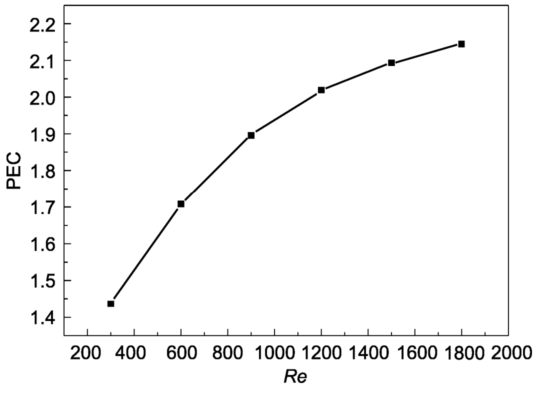


**Figure** 11 Relation between PEC of thin cylinder-interpolated tube and *Re* number.

tube and thin cylinder-interpolated tube. As shown in the figure, the average synergy angle $\beta$ between fluid velocity $\boldsymbol{U}$ and temperature gradient $\nabla T$ in thin cylinder-interpolated tube is also smaller than that in bare tube. Therefore, it can be inferred from eq. (3) that heat transfer will be enhanced between the fluid and the tube wall. Figure 5 shows the relation between *Re* number and average synergy angle $\gamma$ for bare tube and thin cylinder-interpolated tube. As shown in the figure, average synergy angle $\gamma$ between temperature gradient $\nabla T$ and velocity gradient $\nabla u$ in thin cylinder-interpolated tube is bigger than that in bare tube, therefore, the performance of heat transfer unit will be improved by interpolating thin cylinder for disturbing fluid.

Figure 6 shows the relation between *Re* number and average synergy angle $\phi$ for bare tube and thin cylinder-interpolated tube. As shown in the figure, average synergy angle $\phi$ between pressure gradient $\nabla p$ and velocity gradient $\nabla u$ in thin cylinder-interpolated tube is smaller than that in bare tube, which indicates that disturbing fluid leads to a remarkable increase in viscous dissipation and results in an obvious increase in flow resistance. Figure 7 shows the relation between *Re* number and average synergy angle $\theta$ for bare tube and thin cylinder-interpolated tube. As shown in the figure, average synergy angle $\theta$ in the range of calculated *Re* number is bigger than 50°, which shows that the direction of velocity $\boldsymbol{U}$ deviates greatly from the direction of pressure gradient $\nabla p$, and flow resistance increases remarkably. Therefore, it is necessary to keep a better synergy between vectors $\boldsymbol{U}$ and $\nabla p$ for designing lower-resistance heat exchanger.

Figure 8 shows the relation between *Re* number and *Nu* number of fluid in bare tube and thin cylinder-interpolated tube. As shown in the figure, the fluid *Nu* number of thin cylinder-interpolated tube is about 1.8—3 times bigger than that of bare tube, which indicates that heat transfer between the fluid and the tube wall is enhanced after interpolating thin cylinders. Figure 9 shows the relation between *Re* number and fluid resistance coefficient $f$ in bare tube and thin cylinder-interpolated tube. As shown in the figure, the fluid resistance coefficient of thin cylinder-interpolated tube is about 2.2—2.6 times bigger than that of bare tube.

The correlation of fluid resistance coefficient and pressure drop $\nabla p$ can be expressed as

$$\Delta p = f\,\frac{L}{H}\cdot\frac{\rho u_m^2}{2}. \tag{26}$$

According to eq. (26), the relation between *Re* number and fluid pressure drop $\nabla p$ in bare tube and thin cylinder-interpolated tube is shown in Figure 10. It can be observed from the figure that, flow resistance and *Nu* number increase simultaneously with the increase of *Re* number, but the increase amplitude of enhanced heat transfer is close to or greater than that of fluid pressure drop in higher *Re* number.

Figure 11 indicates the relation between *Re* number and PEC value of thin cylinder-interpolated tube. In the range of *Re* number in the figure, PEC value of thin cylinder-interpolated tube is about 1.4—2.2. Even though PEC value increases with the increase of *Re* number, the effect of heat transfer enhancement is not very ideal in this case. However it can be predicted that, under the guidance of the principle of physical quantity synergy, a heat transfer unit with high PEC value can be designed, and as a result, a heat exchanger with high heat transfer coefficient and low flow resistance can be developed.

## 3 Conclusions

(1) Synergetic regulation among physical quantities in the laminar flow field of convective heat transfer is established, and basic characteristics of flow and heat transfer can be reflected by synergy angle $\alpha$, $\beta$, $\gamma$, $\phi$, $\theta$ and $\psi$. The smaller the synergy angle $\beta$ is, the bigger the heat transfer coefficient $h$ will be. This means convective heat transfer between fluid and tube wall will be stronger. The smaller the synergy angle $\theta$ is, the smaller the fluid resistance coefficient $f$ will be. This means fluid pressure drop will be smaller. The bigger the synergy angle $\gamma$ is, the higher the PEC value will be. This means comprehensive performance of a heat transfer unit will be better.

(2) In the optimum design of a heat exchanger, regulating synergy angle $\theta$ for reducing flow resistance plays the same role as improving synergy angle $\beta$ for enhancing heat transfer. If the synergy between velocity $\boldsymbol{U}$ and temperature gradient $\nabla T$ is better, heat can be transferred more, and heat transfer performance of a heat exchanger will be better. If the synergy between velocity $\boldsymbol{U}$ and pressure gradient $\nabla p$ is better, fluid pressure drop and power dissipation of a heat exchanger will be smaller. Therefore, synergy angles $\beta$ and $\theta$ can reflect the relation between mass flux and heat flux, and mass flux and driving force respectively.

(3) According to the principle of physical quantity synergy, it is possible to design excellent heat transfer surfaces and structures ensuring that the increased amplitude of convective heat transfer is close to or bigger than that of flow resistance by improving synergetic relation among physical quantities in the nonisothermal laminar flow field. So the comprehensive performance of a heat exchanger can be improved.


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