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The approach of minimum heat consumption and its applications in convective heat transfer optimization

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ABSTRACT

Based on the analysis of the irreversibility of heat transfer process, a physical quantity called enerty is introduced to establish an equilibrium equation which is different from conventional energy conservation equation. Comparing to minimum entropy generation and minimum entransy dissipation, an approach of minimum heat consumption is proposed to optimize heat transfer process. The transport efficiency, a criterion to evaluate heat transfer performance, is proposed based on the definition of heat consumption rate. By setting minimum heat consumption as optimization objective and fluid power consumption as constraint condition, a momentum equation with additional volume force is constructed through functional variation to numerically simulate convective heat transfer in coupling with energy equation. The results disclose that longitudinal swirl flow with single vortex or multi-vortexes appears in the flow field, which leads to heat transfer enhancement. Additionally, a sub-area method is developed to enhance convective heat transfer characterized by relatively higher *Nu* number, which can be applied to optimize flow field in a circular tube and guide the design for tube insert. The results further indicate that the area ratio and the intensity of additional volume force affect the distribution of temperature and velocity fields. A comparison among minimum heat consumption (MHC), minimum entransy dissipation (MED) and minimum power consumption (MPC) is conducted. The results show that heat transfer performances are close to each other for using the three approaches, which indicates that the proposed MHC approach is effective in designing tube insert or heat transfer unit.

1. Introduction

Requirements of energy- and material-saving, as well as emission reduction in industry, have led to deep attentions to the development for more efficient heat exchangers. For several decades, many efforts have been made to enhance heat transfer or minimize heat exchanger size by disturbing fluid or increasing heat transfer area, etc. [1–8]. However, these work cannot serve as a general guide to optimize heat transfer process. Therefore, it is of great significance to develop more effective approach for heat transfer optimization.

Since entropy or entropy generation can be considered as a measurement of irreversibility in heat transfer process, the existing optimization theory is focused on entropy generation minimization in many applications [9,10]. Bejan proposed the constructal theory to analyze the conducting path for process optimization [11]. Guo et al. proposed the principle of entransy dissipation extremum based on a physical quantity called entransy from the analogy between heat conduction and electrical conduction [12]. They deduced the concept theoretically, and validated it through numerical modeling [13–20]. Liang et al. introduced the entransy-dissipation-based thermal resistance and applied it to the optimization in heat exchanger [21,22]. Besides, Liang et al. also proposed the entransy loss in thermodynamics system and made some optimization analysis in thermodynamics system [23,24]. Chen et al. [25–29] optimized the constructal problems of variable cross-section channel, disc-to-point, area-point and rectangular unit by taking minimum entransy dissipation rate as optimization target. Based on Guo's theory, Liu et al. [30,31] presented an approach of minimum power consumption, and found that it is suitable to convective heat transfer optimization. Besides, as both entropy and entransy equilibrium equations emphasize the quality of energy rather than the quantity of energy, it is significant to introduce a physical quantity that can reflect both quality and quantity of energy, and develop an approach of minimizing irreversible loss to optimize heat transfer process.

2. Theoretical modeling

2.1. Irreversibility of thermal transport process

From heat transfer textbook, the energy conservation equation in the laminar flow for the incompressible fluid can be written as

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Nomenclature

A, B, C_0	Lagrange multipliers
c_p	specific heat capacity, J/(kg K)
C_φ	intensity coefficient
e	enerty, J/kg
E_d	unavailable heat, J/m^3
g	Gibbs free enthalpy, J/kg
h	enthalpy, J/kg
J	functional
Nu	Nusselt number
p	pressure, Pa
Q	heat flow rate, W
Q_d	heat consumption rate, W
$\dot{Q}'''$	inner heat source, W/m^3
$\mathbf{q}$	heat flux vector, W/m^2
Re	Reynolds number
s	entropy, J/(kg K)
T	temperature, K
t	time, s
$\mathbf{U}$	velocity vector, m/s
V	volume, m^3
z	entransy, J K/kg

Greek symbols

λ	heat conductivity, W/(m K)
ρ	fluid density, kg/m^3
μ	viscosity coefficient, kg/(m s)
Φ	heat dissipation from fluid viscosity, W/m^3
Ω	control volume, m^3
Ψ	dissipation function, W/m^3
η_t	transport efficiency of thermal energy

Subscripts

0	reference point, environmental state
1	state 1
2	state 2
x	exergy
e	effective

$$\rho c_p \frac{DT}{Dt} = \lambda \nabla^2 T + \Phi + \dot{Q}''', \tag{1}$$

where ρ is the fluid density, c_p is the specific heat capacity of the fluid, T is the fluid temperature, t is the time, Φ is the viscosity dissipation heat, and $\dot{Q}'''$ is the inner heat source.

Eq. (1) can also be rewritten as

$$\rho \frac{Dh}{Dt} = -\nabla \cdot \mathbf{q} + \Phi + \dot{Q}''', \tag{2}$$

where h is the enthalpy of the fluid, and $-\nabla \cdot \mathbf{q}$ is the heat diffusion of the fluid.

From thermodynamics textbook, in addition, the equilibrium equation for the incompressible fluid can be expressed in terms of entropy,

$$\rho \frac{Ds}{Dt} = -\nabla \cdot \left(\frac{\mathbf{q}}{T}\right) + \frac{\lambda (\nabla T)^2}{T^2} + \frac{\Phi}{T} + \frac{\dot{Q}'''}{T}, \tag{3}$$

where s is the entropy of the fluid, $-\nabla \cdot (\frac{\mathbf{q}}{T})$ is the net entropy flux transferring into and out of fluid elementary volume, $\frac{\lambda(\nabla T)^2}{T^2}$ is the entropy generation rate due to irreversibility of heat transfer process, $\frac{\Phi}{T}$ is the analogical entropy flow induced by viscous dissipation loss, and $\frac{\dot{Q}'''}{T}$ is the entropy flow induced by the inner heat source.

Multiplying both sides of Eq. (3) by T^2, the equilibrium equation is then rewritten in term of entransy [12,30],

$$\rho \frac{Dz}{Dt} = -\nabla \cdot (\mathbf{q}T) - \lambda (\nabla T)^2 + \Phi T + \dot{Q}''' T, \tag{4}$$

where z is the entransy of the fluid, $-\nabla \cdot (\mathbf{q}T)$ is the net entransy flux transferring into and out of fluid elementary volume, $\lambda(\nabla T)^2$ is the entransy dissipation rate due to irreversibility of heat transfer process, ΦT is the analogical entransy flow induced by heat dissipation from fluid mechanical energy, and $\dot{Q}''' T$ is the entransy flow induced by the inner heat source.

The Eqs. (3) and (4) satisfy the second law of thermodynamics. When entropy changes during a heat transfer process, entropy flux is transferred via the fluid, resulting in entropy generation. When entransy changes during a heat transfer process, entransy flux is transferred via the fluid, leading to entransy dissipation. Therefore, entropy generation and entransy dissipation are the trace of process irreversibility. The larger the entropy generation or entransy dissipation rate is, the stronger the process irreversibility will be.

Analyzing Eqs. (2)–(4), we speculate that there should be a physical quantity which may demonstrate the characteristics of energy both in quantity and quality.

Thus we consider a new variable called enerty [32],

$$e = h - T_0 s, \tag{5}$$

and its differential form,

$$de = dh - T_0 ds. \tag{6}$$

The variable e is similar to Gibbs free enthalpy at a reference temperature. If T_0 is the environmental temperature, then e represents the available potential of the fluid, $T_0 s$ denotes unavailable potential of the fluid. The smaller the temperature difference between the fluid and environment is, the lower the available potential of the fluid will be. According to the nature of Gibbs free enthalpy, enerty can be regarded as a state variable.

From the definition in Eq. (5), enerty equals Gibbs free enthalpy when the fluid temperature arrives at the environmental temperature T_0,

$$e_0 = g_0 = h_0 - T_0 s_0, \tag{7}$$

where g_0 is Gibbs free enthalpy at environmental temperature T_0 .

From Eqs. (5) and (7), we have

$$e_x = e - e_0 = (h - h_0) - T_0(s - s_0) = \Delta h - T_0 \Delta s, \tag{8}$$

where e_x is the fluid exergy representing the maximum capacity of the fluid in heat or work transfer, and $T_0\Delta s$ is the fluid anergy representing the unavailable energy in irreversible heat transfer process. Because e is a continuous variable, enerty difference can be expressed as $\Delta e = e_2 - e_1$. Thus, we can see that enerty difference equals to exergy if taking the environment as a reference. Therefore, exergy and enerty are different concepts. Exergy is available energy which can be represented by an "area" in the T-s chart of thermodynamics. Enerty is available potential represented by a "point" in the thermodynamics potential chart.

Fig. 1 demonstrates the change of fluid potential in terms of enerty and enthalpy, which shows the relation between the available and unavailable potentials. From the figure, we can see that it is impossible for available potential to change from state 1 to 2′, as entropy increases in the process, which indicates that the decrease of available potential is charactered by entropy increasing. As a matter of fact, only part of available potential can be effectively

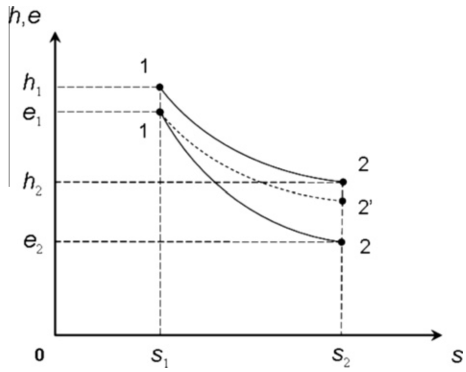


Fig. 1. The relation of potential change between enerty and enthalpy of the fluid.

utilized in a heat transfer or thermodynamic process, because of the irreversible loss.

From the definition of enerty and Eq. (6), we have

$$\frac{De}{Dt} = \frac{Dh}{Dt} - T_0\frac{Ds}{Dt}. \tag{9}$$

From Eqs. (2), (3), and (9), we have

$$\rho\frac{De}{Dt} = -\nabla\cdot\mathbf{q} + \nabla\left(\frac{T_0\mathbf{q}}{T}\right) - \frac{T_0\lambda(\nabla T)^2}{T^2} + \left(1-\frac{T_0}{T}\right)\Phi + \left(1-\frac{T_0}{T}\right)\dot{Q}'''. \tag{10}$$

When the fluid temperature is close to environmental temperature T_0, we then have

$$\left(\rho\frac{De}{Dt}\right)_{T\to T_0} = -\left(\frac{\lambda(\nabla T)^2}{T}\right)_{T\to T_0}, \tag{11}$$

where $(\lambda(\nabla T)^2/T)_{T\to T_0}$ is the trace of process irreversibility, when $T \to T_0$. As the irreversibility always exists in any heat transfer process, an equilibrium equation in terms of enerty can be reasonably obtained as

$$\rho\frac{De}{Dt} = -\nabla\cdot\mathbf{q} - \frac{\lambda(\nabla T)^2}{T} + \Phi + \dot{Q}''', \tag{12}$$

where $\lambda(\nabla T)^2/T$ is the unavailable heat induced by irreversible dissipation loss, which has the unit of energy per cube meter, and can be defined as the heat consumption of the fluid. Obviously, the larger the heat consumption is, the more irreversible the heat transfer process will be. Although temperature gradient ∇T on the right hand side of Eq. (12) may be positive or negative, the term $\lambda(\nabla T)^2/T$ is always positive. This implies that for a system with higher temperature surrounded by its environment with lower temperature, the irreversible process always varies towards the direction of enerty decreasing due to the increase in heat consumption. This statement may be considered as an expression for the second law of thermodynamics differing from that in the classic thermodynamics.

In addition, Eq. (12) can also be written as

$$\rho\frac{De}{Dt} = -\nabla\cdot\mathbf{q} + \Psi + \Phi + \dot{Q}''', \tag{13}$$

where $\Psi = \mathbf{q}\cdot\frac{\nabla T}{T} = \mathbf{q}\cdot\nabla(\ln T)$ is defined as a dissipation function, and the value of dot multiply between thermodynamics flow and force represents the quantification of process irreversibility. If let $E_d = \frac{\lambda(\nabla T)^2}{T}$, then we have

$$\rho\frac{De}{Dt} = \lambda\nabla^2 T - E_d + \Phi + \dot{Q}''', \tag{14}$$

where E_d is the unavailable heat consumed in heat transfer process. Thus, the goal of optimizing a heat transfer process is to minimize E_d as much as possible.

The state variables, i.e., enthalpy, entropy, entransy and enerty, have different meanings, which are indicated by Eqs. (2)–(4), and (12). It can be seen that enthalpy reflects the quantity of energy, while entropy and entransy reflect the quality of energy. Whereas, enerty with the unit same as energy can represent the energy potential of the fluid both in quantity and quality. Furthermore, for a thermodynamic or heat transfer process, although there are strictly definitions for entropy and entransy, it is difficult to understand them either in the concept or in the unit. However, enerty has a clear meaning which is well explained by its unit and physical characteristic. So enerty, as a new state variable, is more intuitional than other state variables.

For the incompressible fluid, the state variables s, e and z as well as their corresponding differentials are listed in Tab. 1. The transfer quantities include entropy flux $\frac{\mathbf{q}}{T}$, heat flux $\mathbf{q}$ and entransy flux $\mathbf{q}T$, and the irreversible quantities include entropy generation rate $\frac{\lambda(\nabla T)^2}{T^2}$, heat consumption rate $\frac{\lambda(\nabla T)^2}{T}$ and entransy dissipation rate $\lambda(\nabla T)^2$ [12]. Obviously, very good symmetric relation has been revealed in Table 1.

From the definition of heat consumption rate, it contains both temperature and temperature gradient. When a system is in balance at relatively higher temperature and its temperature is more uniform, the unavailable heat will be smaller. Therefore, it is reasonable to use heat consumption as an optimizing objective in heat transfer process optimization. So, for convective heat transfer in a tube, if we can obtain relatively higher and more uniform fluid temperature, the thermal resistance between the tube wall and the fluid will be smaller. In this circumstance, convective heat transfer will be enhanced greatly.

Thus, apart from optimizations based on minimum entropy generation and minimum entransy dissipation, minimum heat consumption is regarded as a basic approach of process optimization, which can be stated as the principle of minimum dissipation in general. They are different from minimum power consumption in nature. The formers indicate the irreversible dissipation loss, the latter denotes the power consumed in the process.

2.2. Transport efficiency of thermal energy

Compared to mechanical, electric or chemical energy, thermal energy is a kind of low grade energy, which cannot be fully utilized, and only part of thermal energy is effectively transferred in a system. So, it is crucial to minimize unusable thermal energy by decreasing heat consumption as much as possible.

For a heat transfer problem, transport efficiency of thermal energy is usually not considered. Take stable convective heat transfer in a tube as an example. The heat transferred through tube wall is carried away by the fluid. According to the first law of thermodynamics, the energy is in balance. So it is not necessary to evaluate transport efficiency in this case. However, from the view of the second law of thermodynamics, if boundary heat flux and fluid inlet temperature keep constant, fluid outlet temperature will represent the quality of thermal energy carried by the fluid. Although the total amount of thermal energy does not change, its quality changes with different manner of enhanced heat transfer under the same boundary and inlet conditions. The higher the fluid outlet temperature is, the higher the grade of thermal energy will be. This is to say that the thermal energy carried by the higher temperature fluid is of higher utilization efficiency than that carried by the lower temperature fluid. According to the second law of thermodynamics, therefore, the higher the grade of thermal energy is, the better the availability of thermal energy will be. Thereby the transport efficiency will be higher.

Through integral in the whole fluid region, the total heat consumption rate is given as

Table 1
The relations among physical quantities entropy, enerty and entransy as well as entropy generation, heat consumption and entransy dissipation.

Variable/unit	Transport quantity	Irreversible nature	Variable differential
s J/(kg K)	$\frac{\mathbf{q}}{\mathbf{T}}$	$\frac{\lambda(\nabla T)^2}{T^2}$	$ds = c\frac{dT}{T}$
$e = h - T_0 s$ J/kg	$\boldsymbol{q}$	$\frac{\lambda(\nabla T)^2}{T}$	$de = cdT - T_0 ds$
z (J K)/kg	$\boldsymbol{q}T$	$\lambda(\nabla T)^2$	$dz = cTdT$

$$Q_d = \iiint_\Omega E_d dV = \iiint_\Omega \frac{\lambda(\nabla T)^2}{T} dV \tag{15}$$

Then transport efficiency of thermal energy can be defined as

$$\eta_t = \frac{Q_e}{Q} = \frac{Q - Q_d}{Q} = 1 - \frac{Q_d}{Q}, \tag{16}$$

where Q is the total heat flow rate transferred into or out of the system, Q_e is the effective heat flow rate. It is obvious that when the fluid has more uniform temperature and higher average temperature, its heat transfer ability will be better due to smaller heat consumption, i.e., the smaller irreversible loss Q_d will lead to the higher transport efficiency in a heat transfer process. Therefore, the transport efficiency of thermal energy can be used as a criterion to evaluate the performance of a heat transfer unit or device.

2.3. Heat transfer process optimization

Taking minimum heat consumption as optimization objective and fixed power consumption as constrain condition for a convective heat transfer process, the Lagrange functional can be obtained as follows [15,30].

$$J = \iiint_\Omega \left[\frac{\lambda(\nabla T)}{T} + C_0(\rho\mathbf{U}\cdot\nabla\mathbf{U} - \mu\nabla^2\mathbf{U})\cdot\mathbf{U} + A\nabla\cdot\mathbf{U} + B(\lambda\nabla^2 T - \rho c_p\mathbf{U}\cdot\nabla T)\right]dV, \tag{17}$$

where $(\rho\mathbf{U}\cdot\nabla\mathbf{U} - \mu\nabla^2\mathbf{U})\cdot\mathbf{U} = -\nabla p\cdot\mathbf{U} = W$ is the work consumed by the fluid. C_0, A and B are Lagrange multipliers respectively.

Through functional variation with respect to Lagrange multipliers A and B, we can have mass conservation equation

$$\nabla\cdot\rho\mathbf{U} = 0, \tag{18}$$

and energy conservation equation

$$\rho c_p\mathbf{U}\cdot\nabla T = \lambda\nabla^2 T. \tag{19}$$

Through functional variation with respect to velocity vector $\mathbf{U}$, we can obtain the following optimization equation,

$$\rho[\mathbf{U}\cdot\nabla\mathbf{U} + \mathbf{U}\times(\nabla\times\mathbf{U})] - 2\mu\nabla^2\mathbf{U} - \frac{\nabla A}{C_0} - \frac{\rho c_p B\nabla T}{C_0} = 0. \tag{20}$$

By comparing Eq. (20) with conventional momentum equation, and letting

$$\rho\mathbf{U}\times(\nabla\times\mathbf{U}) = \frac{\nabla A}{C_0} + \nabla p + \mu\nabla^2\mathbf{U}, \tag{21}$$

we can obtain a new momentum equation

$$\rho\mathbf{U}\cdot\nabla\mathbf{U} = -\nabla p + \mu\nabla^2\mathbf{U} + \frac{\rho c_p B}{C_0}\nabla T, \tag{22}$$

where $\frac{\rho c_p B}{C_0}\nabla T$ denotes the additional volume force induced to optimize the flow field, which represents a virtual heat field, and plays a similar role as electric or magnetic field. The coefficient C_0 is related to the intensity of the heat field.

Through functional variation with respect to temperature T, we can obtain the following constrain equation,

$$\rho c_p\mathbf{U}\cdot\nabla B = \frac{2\lambda\nabla^2 T}{T} - \frac{\lambda(\nabla T)^2}{T^2} - \lambda\nabla^2 B. \tag{23}$$

From Eqs. (21) and (23), it can be noticed that the multipliers A and B are associated with pressure p and temperature T respectively. The variables $\boldsymbol{U}$, T, p and B can be solved by the coupling Eqs. (18), (19), (22), and (23) to obtain a solution of convective heat transfer problem.

3. Calculation and analysis

A model of circular tube with geometry sizes $D = 20$ mm, $L1 = 1200$ mm, $L2 = 300$ mm and $L3 = 200$ mm is shown in Fig. 2. Where D is the tube diameter, $L1$ is the length of entrance region, $L2$ is the calculation domain of heat transfer optimization, and $L3$ is the length of stable region. The working fluid is water, and its inlet temperature is 300 K. The tube wall keeps constant temperature, and its temperature is 310 K. Thus, by solving the above governing equations, we can examine the effect of virtual heat field on the flow organization in the tube to validate the theoretical model. Noting that the value of coefficient C_0 is related to the intensity of virtual heat field, we find that the magnitude of C_0 results in different flow field structures. In addition, the value of C_0 is restricted to a certain range which depends on the geometric sizes and boundary conditions.

The CFD software FLUENT 6.3 is used in the calculation. The SIMPLEC algorithm is used for coupling the pressure and velocity. The convection and diffusion terms are discretized by the QUICK scheme. The user defined function (UDF) and user defined scalar (UDS) are utilized to solve the governing Eqs. (22) and (23). The convergent solutions are obtained when the residuals of all governing equations are less than 10^{-8}.

Figs. 3 and 4 show the results for the case in which minimum heat consumption is set as optimization objective, and fixed power consumption as constraint condition. In the calculation, different value of $C_\varphi = \frac{\rho c_p}{C_0}$ is considered under the same Reynolds number 200, which is defined as intensity coefficient. In Fig. 3, it can be seen that when C_ϕ is 0.05, there is longitudinal swirl flow with a single vortex, and when C_ϕ is increased to 0.4, the vortex number becomes eight. So, larger value of C_ϕ means relatively stronger additional volume force, which implies that the fluid disturbance is more intensive, and as the results, heat transfer is enhanced.

According to the principle of heat transfer enhancement based on the fluid in the tube [33], a proper way of enhancing heat transfer is to disturb the fluid in the core area of the tube to achieve uniform fluid temperature in this area, and enlarge fluid temperature gradient near the wall. Then, the entire area of the tube section is separated into two parts: core area (area I) and boundary area (area II). It can be seen that there exists a swirl flow with two vortexes in Fig. 5, four vortexes in Fig. 6, and six vortexes in Fig. 7. The Reynolds number is 200 in the calculation. From Figs. 5–7, it can be noticed that even the coefficient $\rho c_p/C_0$ is chosen as the same, the different area ratio leads to the different flow structure, and the vortex number is increased with the increase of area I.

The impact of intensity coefficient on vortex number is further investigated and ten vortexes are predicted theoretically in Fig. 8.

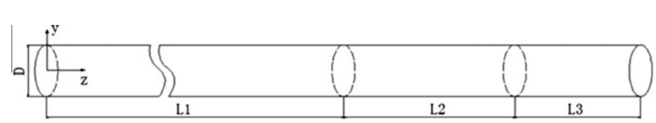
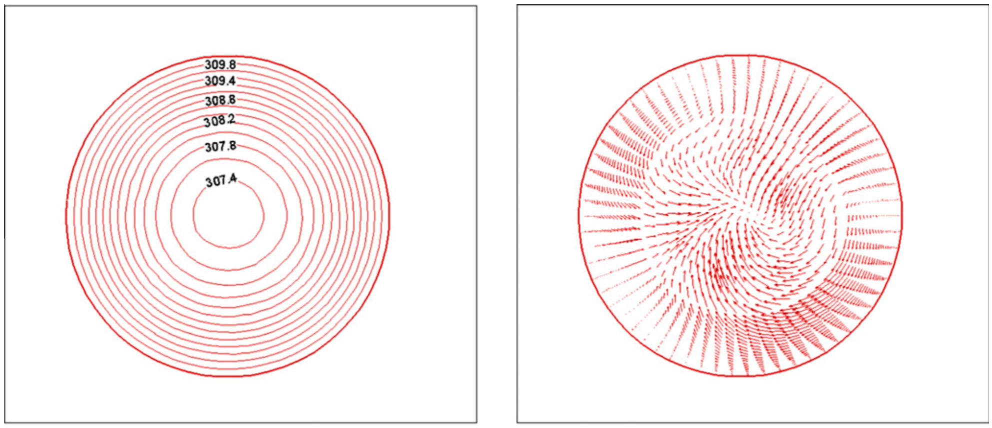
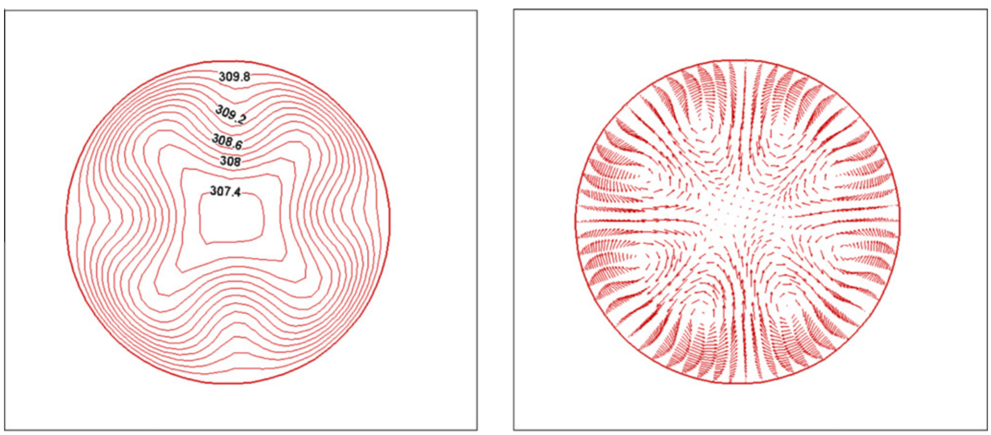


Fig. 2. Schematic of the calculation model.



(a) Temperature field (b) Velocity vector

Fig. 3. Flow field structure with single vortex and temperature field ($\rho c_p/C_0$ = 0.05, Re = 200).

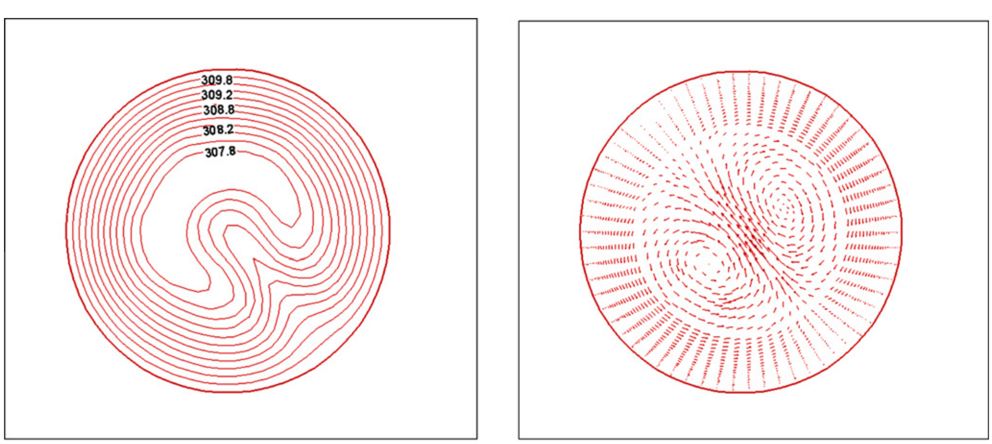


(a) Temperature field (b) Velocity vector

Fig. 4. Flow field structure with eight vortexes and temperature field ($\rho c_p/C_0$ = 0.4, Re = 200).

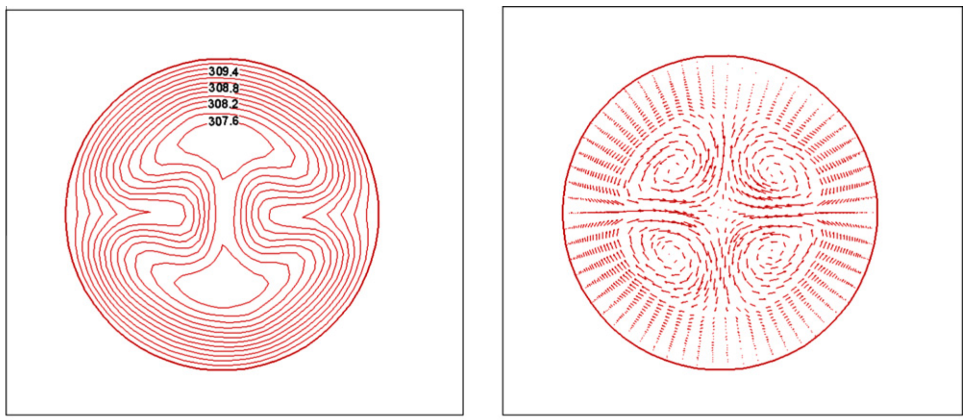
Although the area ratio is the same, the intensity coefficients $\rho c_p/C_0$ are 3 and 0.1 in the Fig. 7 and 12 and 10 in the Fig. 8 respectively. It can be observed again that larger value of $\rho c_p/C_0$ leads to more vortexes and more uniform fluid temperature. Consequently convective heat transfer in the tube is enhanced.

A comparison among approaches of minimum entransy dissipation (MED), minimum heat consumption (MHC) and minimum power consumption (MPC) is conducted. Fig. 9 gives a comparison of heat transfer characteristic and flow resistance for different optimization objectives. The results show that heat transfer capacity



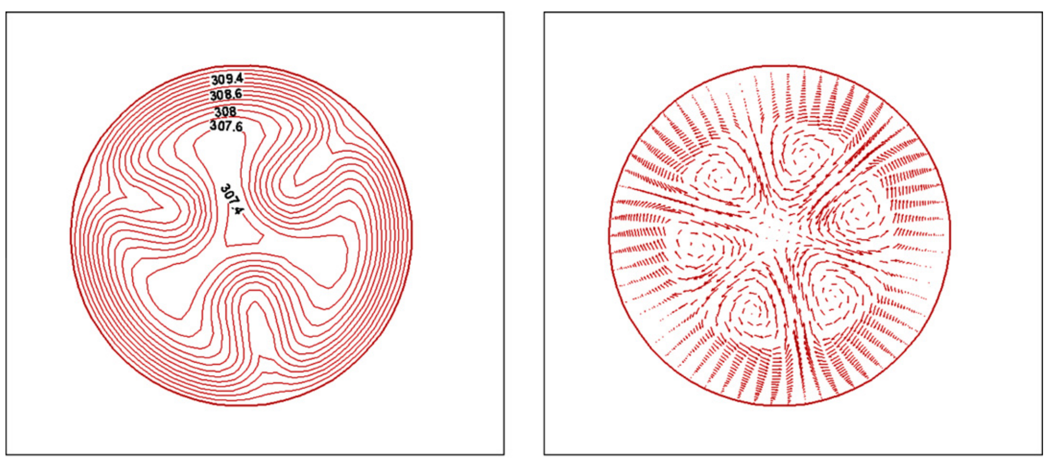
(a) Temperature field (b) Velocity vector

Fig. 5. Flow field structure with two vortexes and temperature field ($[\rho c_p/C_0]_{AreaI=0.5}$ = 3.0, $[\rho c_p/C_0]_{AreaII=0.5}$ = 0.1, Re = 200).



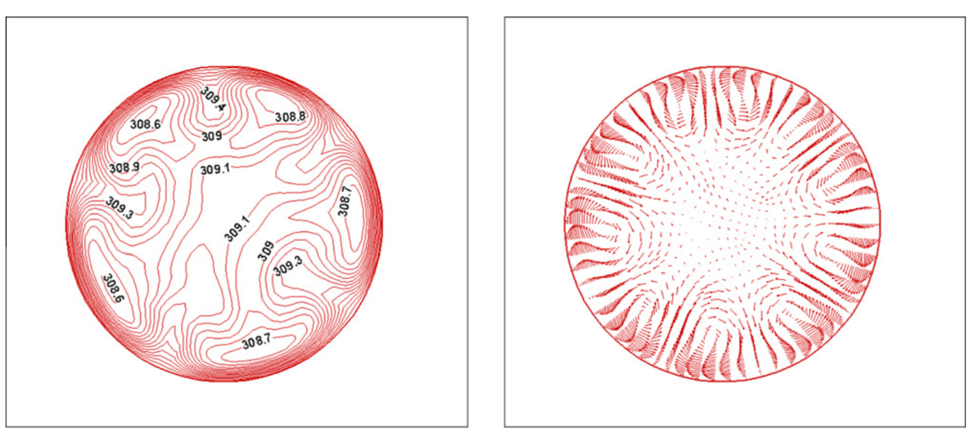
(a) Temperature field (b) Velocity vector

Fig. 6. Flow field structure with four vortexes and temperature field ($[\rho c_p/C_0]_{AreaI=0.6}$ = 3.0, $\rho c_p/C_0]_{AreaII=0.4}$ = 0.1, Re = 200).



(a) Temperature field (b) Velocity vector

Fig. 7. Flow field structure with six vortexes and temperature field ($[\rho c_p/C_0]_{AreaI=0.7}$ = 3.0, $[\rho c_p/C_0]_{AreaII=0.3}$ = 0.1, Re = 200).



(a) Temperature field (b) Velocity vector

Fig. 8. Flow field structure with six vortexes and temperature field (Flow field structure with ten vortexes and temperature field ($[\rho c_p/C_0]_{AreaI=0.7}$ = 12, $[\rho c_p/C_0]_{AreaII=0.3}$ = 10, Re = 200).

predicted by the approach of MPC is the best at the same power consumption, and the MHC lies between the MPC and the MED. Flow resistances predicted by the approaches of MED, MHC and MPC are almost the same.

In order to verify the concept of transport efficiency of thermal energy, convective heat transfer both in smooth and enhanced tubes was simulated. Fig. 10 gives a comparison of transport efficiency between smooth and enhanced tubes, in which wall heat

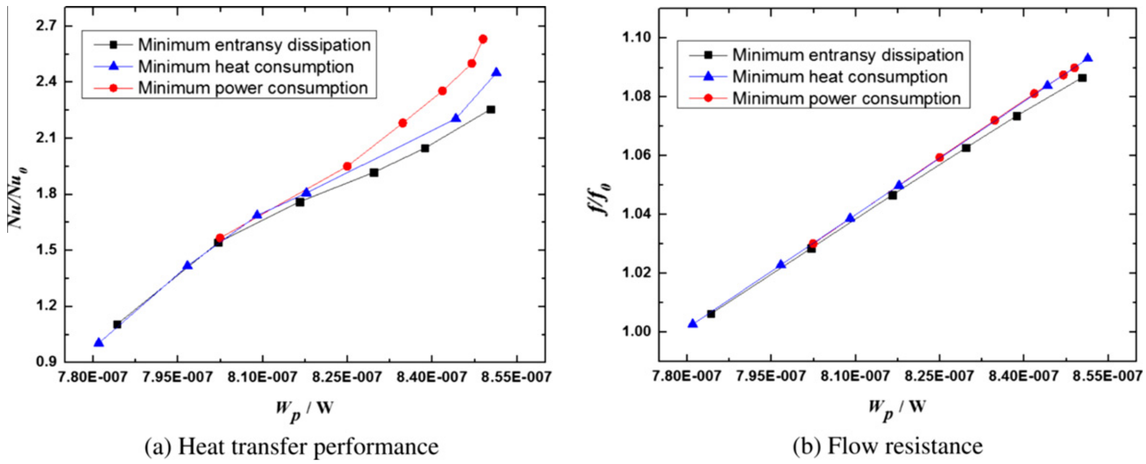


Fig. 9. Heat transfer process optimization under different methods.

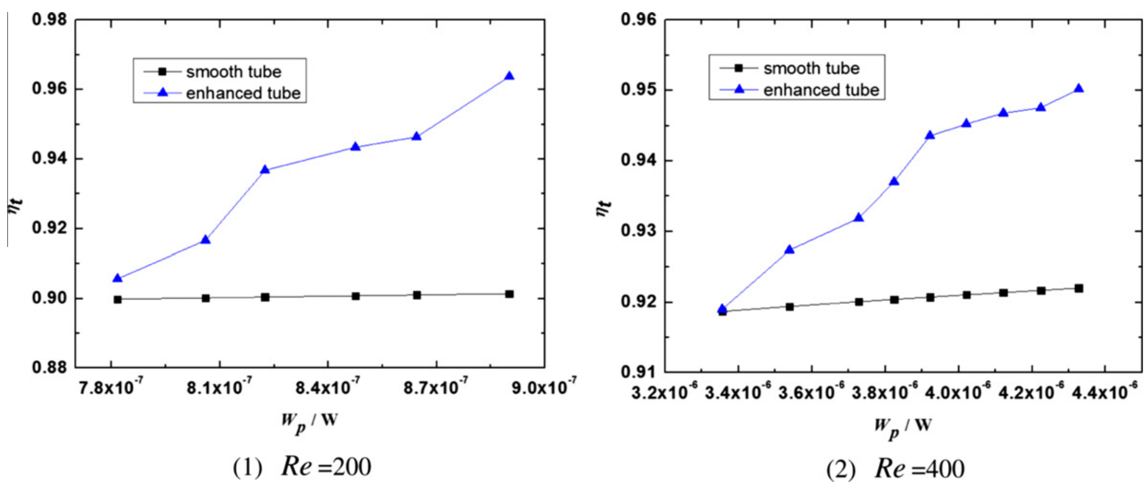


Fig. 10. Transport efficiency of thermal energy of smooth tube and enhanced tube under different power consumption.

flux is 9000 W/m^2, and Reynolds number are 200 and 400 respectively. The results show that if the MHC approach is adopted to realize heat transfer optimization in a circular tube, its transport efficiency will be much higher than the smooth tube. This validates that higher transport efficiency means less irreversible loss at the same power consumption. Compared with the entropy generation minimization [9,10], therefore, the MHC approach is more suitable for the optimization of convective heat transfer, because it gives a way of quantificationally evaluating the performance of transport process.

4. Conclusions

A new physical quantity called enerty is introduced to describe the irreversibility of heat transfer process, and an approach of minimum heat consumption is developed to optimize heat transfer process. Based on the concept of heat consumption rate, the transport efficiency of thermal energy is defined as a criterion to evaluate convective heat transfer performance. In addition, a sub-area method is used in optimization calculation to achieve relatively stronger convective heat transfer with relatively higher Nusselt number, which may guide the design of tube insert or heat transfer unit based on the optimized flow structure in a circular tube.

Stable longitudinal swirl flow with vortexes in the tube is achieved by the approach of minimum heat consumption, which results in heat transfer enhancement owing to the action of additional volume force called heat field effect. Through comparison with approaches of minimum entransy dissipation and minimum power consumption, the effectiveness of optimization approach with minimum heat consumption as optimization objective is identified.

Acknowledgements

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