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Active optimization design theory and method for heat transfer unit and its application on shape design of cylinder in convective heat transfer

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ABSTRACT

When heat transfer enhancing technology is applied to design heat transfer unit, the improvement of heat transfer will usually accompanied with a dramatically increase of flow resistance. The main objective of present paper is to propose an optimization design theory and develop a corresponding approach to design heat transfer unit, in which the heat transfer and flow resistance will be considered simultaneously. As an example, this paper shows how to obtain a high performance tube in convective heat transfer by optimizing the shape profile using the theory and approach. A 2-D model is used in a direct problem solver, which is solved by finite element software and provides the numerical results in the optimization. Power consumption of flow field P_f is used to measure the flow resistance while generalized thermal resistance R_h is used to measure the thermal resistance. Meanwhile, genetic algorithm (GA) and simplified conjugate-gradient method (SCGM) are applied in this study to optimize the objective function composed by these two aspects. Based on the results of numerical calculation, the optimal velocity field with 61.93% decrease in P_f and 17.13% increase in R_h is obtained. Subsequently, the performances at different inlet velocities V_i are investigated. It is found that the shape profile obtained at $V_i = 0.1$ m/s always has a better performance than the circular one, even if it is not the ideal shape. Finally, the further optimizations based on different V_i and objective functions are discussed. The results prove that the proposed optimization method is effective in designing the shape of tube.

1. Introduction

With the development of economy, the requirements of the energy saving have caused widespread concern in industry. In recent decades, numbers of efficient heat exchangers were made to enhance performance [1–3]. Most of these work concentrated on the effects of the structure of the heat transfer units to the convective heat transfer and their comprehensive performance evaluation. However, the parameters of structure in their work are usually obtained by experience. Therefore, there is a lack of general method to optimize parameters in heat transfer process. Thus, it is of great value to develop more optimization approach for structural design of the heat transfer units.

Guo et al. [4] proposed the field synergy principle which reveals the relationship between local behavior and the comprehensive performance of convective heat transfer. Liu et al. [5–7] explained physical quantity synergy principle from field synergy principle by reflecting the physical mechanism of convective heat transfer in the laminar and turbulence flows. According to the field synergy principle, changing the velocity field will influence the heat transfer process significantly. Based on this idea, lots of theoretical work [8,9] are proposed and the heat transfer units with high performance are obtained by optimizing the flow field structure. Their results prove that the flow field with multi-longitudinal vortexes shows excellent heat transfer performance. Meanwhile, it is found that the more the vortexes are, the higher *Nu* number will be. In the industrial applications, the tube with inserts is used to achieve the vortex structures and raise the performance [10–12].

Optimizing the shape of the heat transfer units is another effective method to change the velocity field. For instance, using finned oval tubes will have a better fluid dynamic configuration when comparing with the circular one. It is reasonable to expect a reduction in pressure drop and an increase in heat transfer. Since Brauer [13] reported a survey of experimental results comparing elliptic and circular arrangements, more and more studies are analyzed

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Nomenclature

C	weighting factor
C_P	specific heat, 1.006 kJ/(kg K)
J	objective function
k	thermal conductivity, 2.548×10^{-2} W/(m K)
K	scale coefficient
l	perimeter of initial tube
L	length of flow field, 200 mm
p	pressure, Pa
p_o	outlet pressure, 0 Pa
P_f	power consumption of flow field, W
Q_t	total heat flow
r_i	undetermined coefficients to optimize
r_i'	undetermined coefficients after scaling
Re	Reynolds number, $\rho V_i W/\mu$
R_h	generalized thermal resistance (K/W)
T	air temperature, K
T_L	temperature of the inlet flow, 293.15 K
T_W	temperature of the surface, 353.15 K
u	velocity in x-direction, m/s
v	velocity in y-direction, m/s
V_i	inlet velocity, m/s
$\vec{V}$	velocity vector, $u\vec{i} + v\vec{j}$
x, y, z	rectangular coordinates

Greek symbols

β	search step size
$\dot{\varphi}_h$	entransy dissipation rate
γ	conjugate gradient coefficient
μ	dynamic viscosity, 1.824×10^{-5} N s/m^2
π	search direction
θ_i	polar angle, °
ρ	air density, 1.205 kg/m^3

Superscript

m	iteration number in simplified conjugate-gradient method
n	generation number

based on this idea [14–16]. However, just similar with the problems met in applying the inserts, the ellipse cannot accurately describe the ideal shape of heat transfer units and needs to be further optimized.

Since one of the earliest papers solving the inverse heat transfer problems (IHTP) for simple shapes was reported by Stolz [17] in 1960, IHTP is getting more and more attention by researchers. Up to now, The IHTP is widespread in the fields of aerospace, power engineering, mechanical engineering, constructional engineering and biomedical engineering. Recently, shape design problem becomes an important type of IHTP, which is essential to meet the requirements of industrial equipment. In general, it is difficult to get analytic solution except some very specific cases for IHTP owing to the characteristics of the ill-posedness and nonlinearity. Therefore, numerical methods are used to obtain the numerical solution. Nowadays, there are a number of methods available for IHTP, such as regularization method [18], genetic algorithm [19], and conjugate gradient method [20].

In this paper, two influencing factors on fluid flow and heat transfer performances of the tube are considered simultaneously, of which power consumption of flow field is used to measure the flow resistance and generalized thermal resistance is used to measure the thermal resistance. However, at the certain inlet velocity, there is a universal phenomenon that an increase in the heat transfer coefficient often leads to an increase in the flow resistance, which means heat transfer coefficient and low flow resistance are incompatible in general. Therefore, a weighting constant is used to balance the power consumption and the generalized thermal resistance according to the requirement of design purpose. Furthermore, the genetic algorithm (GA) and simplified conjugate-gradient method (SCGM) [21] are used in this paper to identify IHTP and optimize the shape of the tube. It is necessary to point out that the genetic algorithm can be used not only for the IHTP [22]. It can be also applied for optimizing thermodynamics cycles, such as regenerative Clausius and organic Rankine cycles [23,24].

The aim of present paper is to put forward an optimization design theory and develop a corresponding approach to design high performance heat transfer unit with high heat transfer coefficient and low flow resistance. In order to verify the theory and approach, the convection heat transfer optimization problem of flowing around a circle tube is presented and discussed.

2. Theories and optimization

2.1. Direct problem solver

The two-dimensional model of air flow around the tube is shown in Fig. 1. A heating tube of perimeter l is placed in the flow field and the surface of tube is fixed at a uniform high temperature T_W. Air stream at low temperature T_L flows around and cools down the tube with inlet velocity V_i. The air is regarded as a Newtonian fluid, and its properties are assumed constant. In addition, the flow field is considered to be steady-state. In order to simulate an array of tubes, both the top and the bottom boundaries are under periodic boundary condition. For the flows, the governing equations expressing the laws of conservation in mass, momentum, and energy are:

$$\text{Mass}: \quad \nabla \cdot \vec{V} = 0 \tag{1}$$

Momentum in x-direction :

$$\rho\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = -\frac{\partial p}{\partial x} + \nabla \cdot (\mu \nabla u) \tag{2}$$

Momentum in y-direction :

$$\rho\left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right) = -\frac{\partial p}{\partial y} + \nabla \cdot (\mu \nabla v) \tag{3}$$

$$\text{Energy}: \quad \rho C_P\left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y}\right) = \nabla \cdot (k\nabla T) \tag{4}$$

where the velocity vector and the gradient operator are represented by $\vec{V} = u\vec{i} + v\vec{j}$ and $\nabla \equiv \vec{i}\frac{\partial}{\partial x} + \vec{j}\frac{\partial}{\partial y}$, respectively. These boundary conditions are described as follows:

(1) at $x = 0$: $u = V_i$, $v = 0$, $T = T_L$;
(2) at $x = L$: $\frac{\partial T}{\partial x} = 0$, $p = p_o$;
(3) at surface of the tube: $\vec{V} = 0$, $T = T_W$;
(4) at $y = 0$ and $y = W$: $\vec{V}_{source} = \vec{V}_{dest}$, $p_{source} = p_{dest}$, $-(k\nabla T)p_{dest} = (k\nabla T)_{source}$, $T_{dest} = T_{source}$.

The above governing equations along with the boundary conditions are solved by adopting the finite-element method. In this