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Numerical study on heat transfer enhancement of circular tube bank fin heat exchanger with interrupted annular groove fin

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HIGHLIGHTS

- Conjugate heat transfer performance of an IHAG fin-tube heat exchanger was studied.
- IHAG fin has dual efficacy of fluid flow guiding and detached eddy inhibition.
- IHAG fin could not efficiently enhance heat transfer at lower Reynolds numbers.
- IHAG fin has an excellent performance at higher Reynolds numbers.
- The studied locations of IHAG have fairly limited effects on average characteristics.

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ABSTRACT

A variation in fin surface geometry is an effective approach to improve streamline pattern when fluid flows through the channel form by circular tube bank fins. The structure of interrupted half annular groove (IHAG) fin is different from commonly used fin patterns. In this paper, a conjugate heat transfer numerical method is employed to investigate the average heat transfer and fluid flow characteristics of the staggered circular tube bank fin heat exchanger with IHAG fin. The reference fin is the plain fin with a corresponding configuration. The annular groove's radial and circumferential locations are the main parameters to investigate. The results reveal that (1) The interrupted annular groove has dual efficacy of fluid flow guiding and detached eddy inhibition to reduce the size of wake region; (2) At lower Reynolds numbers, the interrupted annular groove fin surface could not efficiently enhance heat transfer under identical pumping power criteria, and the excellent performance of the interrupted annular groove fin can be achieved at higher Reynolds numbers. There is an average 35% increase in the friction factor, while for Reynolds number ranged from 600 to 2500, the average Nusselt number is increased by 10%–40%, and the corresponding thermal performance factor ranges from 7% to 27%; (3) The studied annular groove's radial and circumferential locations have a fairly limited effect on the average heat transfer and fluid flow characteristics.

1. Introduction

Tube bank fin heat exchangers are widely used in many industrial applications such as vehicle radiators, air conditioning systems and heaters. The flow structure in a channel formed by tube bank fins is significantly affected by the tube's configuration. The form drag can be reduced by using flat tubes, but the welding technology as used to join the fins to the tubes would result in energy consumption and environmental pollution. Taking some measures to realize energy saving and emission reduction in tube-fin joining process is therefore very necessary. It is an effective measure to replace flat tubes with circular ones because tube expansion is the process which joins the fins to the tubes and the circular tube fin heat exchangers have better machining property.

When the working fluid flows through the circular tube, however, the streamline pattern is worsened and the wake region would be expanded. These disadvantages would lead to a remarkable increasing of pressure-drop penalty when the flow passes the front stagnation point of the tube and the wake region, and the wake flow would worsen the heat transfer rate also. An effective approach to overcome the disadvantages of this type of exchanger is to improve flow field structures through variations in fin surface

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Nomenclature

A_o	total surface area [m^2]
A_{front}	inlet cross section area [m^2]
c_p	specific heat capacity [kJ/(kg K)]
C_λ	ratio of thermal conductivity $C_\lambda = \lambda_f/\lambda_a$ [−]
D	tube outer diameter [m]
D_g	groove base arc diameter [m]
f	friction factor: $f = 2\Delta P/L$ [−]
h_o	average heat transfer coefficient [W/(m^2 K)]
L	dimensionless length of fin [−]
Nu	Nusselt number: $Nu = h_o D/\lambda$ [−]
P	dimensionless pressure [−]
Pr	Prandtl number: $Pr = \eta c_p/\lambda$ [−]
Q	heat transfer rate [W]
Re	Reynolds number: $Re = \rho u_{in} D/\eta$ [−]
r_g	annular groove inner radius [m]
S_1	transversal pitch between circular tubes [m]
S_2	longitudinal pitch between circular tubes [m]
T_p	fin spacing [m]
T	temperature [K]
U_i	components of dimensionless velocity vector [−]
X_i	dimensionless coordinates axes [−]

Greeks

δ	thickness of fin, tube and annular grove [m]
λ	thermal conductivity [W/(m K)]
η	viscosity [Pa s]
ε	relative error [−]
β	angle between the adjacent end-cross sections [°]
θ	angle between the two end-cross sections [°]
Θ	dimensionless temperature [-]
ΔP	dimensionless pressure drop [−]

Subscripts

a	air
ave	average
f	fin
in	inlet
I, II	fin surface I and II
out	outlet
ref	reference
t	tube
tube$i$	the ith tube
w	wall

Superscripts

f	fin

geometry. And numerous modified fin patterns have been subsequently developed and widely investigated in the last two decades, mainly including the wavy fin, the slit fin, the louvered fin, the fin with punched vortex generators (VGs), and some combination enhanced fins [1–7].

Traditional understanding of heat transfer enhancement mechanism of the enhanced fin patterns has mainly involved extending the surface area, increasing the length of flow pathlines, interrupting the development of flow boundary layer by repeated geometrics, reducing the thickness of the thermal boundary layers, or generating swirl and flow destabilization. A modified fin pattern called as the circular tube with interrupted annular groove fin is different from the above mentioned fins both in structure and heat transfer enhancement mechanism. For example, the wavy fin can not only extend the heat transfer area and lengthen the flow pathlines, but also provide better air flow mixing when fluid flows over it. The slit fin and louvered fin with the interrupted geometries can decrease the thermal boundary layer thickness and enhance fluid mixing through large-scale instabilities. The fin with punched VGs can generate longitudinal vortices around it and the vortices would spread along main stream direction. The punched VGs in fluid field not only interrupts the development of thermal boundary layer, but also generates mainly longitudinal vortices and flow destabilization to enhance moment and mass transfer of fluid between near the wall and far away from the wall. While for interrupted annular groove fin, there are four sections of annular grooves around each tube, as shown in Fig. 1. It is expected that the interrupted annular groove could interrupt the development of boundary layer, guide the fluid flowing around tube to obtain the better streamline pattern, and inhibit the detached eddy to reduce the wake region. Thus, a better heat transfer performance of this fin pattern may be achieved. To the authors' certain knowledge, the experimental and numerical investigates of the heat transfer of a tube bank with interrupted annular groove fin element is scarcely reported. Thus, the main object of this paper is to investigate its heat transfer performance and to promote understanding of the heat transfer enhancement mechanism of this enhanced fin pattern.

There are two methods to investigate the heat transfer performance of a fin pattern. One is experimental method, and the other is numerical method. Due to a wide range of geometric variations of the enhanced fin pattern, by the experiment method to obtain multi-parameter effects on the air-side performance of this type of exchanger is very complex and expensive [1,2,8–12]. Numerical method is a powerful research tool to alleviate the effort of doing experiments. Recently, much more accurate numerical models have been developed to study the existing wavy fin, slit fin, louvered fin, punched VGs fin, and other fin with surface modifications [3–7,13–16]. Hsieh and Jang [13] numerically investigated the effects of louver fin's multi-parameter on fin side performance and then optimized it's parameters by the Taguchi method. He et al. [14] investigated the conjugate heat transfer in a circular tube bank fin

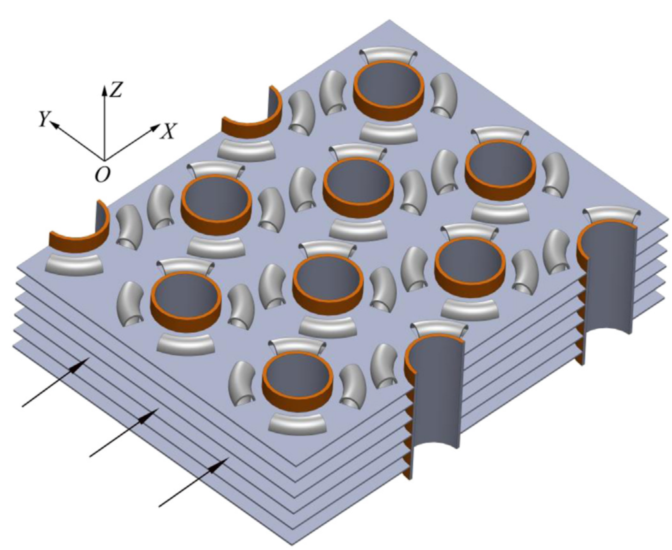

Fig. 1. Schematic view of circular bank fin heat exchanger with interrupted annular groove fins.