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Analysis on heat transfer and pressure drop of fin-and-oval-tube heat exchangers with tear-drop delta vortex generators

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ABSTRACT

Numerical simulations are performed to examine the thermal and flow characteristics of tear-drop delta vortex generators (VGs) on the fin surfaces of fin-and-oval-tube heat exchangers. The VGs deployed in a "common flow up" configuration behind tubes can induce longitudinal vortices and reduce the area of wake region, which results in significant heat transfer enhancement and negligible augmentation of pressure drop compared with plain fins without VGs. The heat transfer performance and pressure loss are analyzed using the dimensionless parameters j/j_0, f/f_0, and $R = (j/j_0)/(f/f_0)^{1/3}$ with Re_{Dc} ranging from 255 to 1533. The results indicate that the tear-drop delta VGs have a better thermal-hydraulic performance than plane delta VGs and the value of R reaches as high as 1.06–1.23 at the ratio of division for chord length (l_1/l_2) being 2/3. Further parameters study reveals that when the height of VGs is 0.6 times of fin pitch, the lateral length is 0.3 times of fin pitch, and the chord length is 1.2–1.4 times of fin pitch, the optimal overall performance can be obtained. The mechanism of heat transfer enhancement is investigated by intensity of secondary flow and field synergy principle. It suggests that for the case with higher Nusselt number, the corresponding secondary flow intensity is higher and the synergy angle is smaller. In addition, for each case, where the Nusselt number is higher, the secondary flow intensity is higher and the synergy angle is smaller.

1. Introduction

Heat transfer enhancement is of great significance in modern industrial process. Achieving higher heat transfer rates contributes to substantial energy saving and lower greenhouse gas emissions. The passive methods to increase heat transfer performance, including applying fluid additives or special surface geometries, are extremely appealing for the lack of external power. For the former method, Sheikholeslami and his coworkers carried out extensive investigations on using nanofluid to improve heat transfer performance. The nanofluid could be water based with CuO [1,2] or Fe_3O_4-Ethylene glycol [3], and its heat transfer performance in a porous medium was simulated. They also examined the influence of CuO nanoparticles on the heat transfer characteristics of PCM in solidification process [4,5] and found that higher heat transfer rate was achieved compared with pure PCM. In addition, they performed the experiments on the condensation process for nano-refrigerant of R600a with CuO as nanoparticles [6,7] and reported that the nanoparticles could intensify the conduction of the tube surface and micro-convection in the fluid. For the later method, the special surface geometries are usually applied in the air-to-water heat transfer process due to the high thermal resistance in the air side. Sheikholeslami et al. studied the thermal and flow characteristics of a water to air heat exchanger with helical turbulators [8] and perforated circular-ring [9]. The corrugated fins [10] or slotted fins [11] are also typical surface modifications to increase the heat transfer area and strengthen the flow disturbance. The main drawback of implementing such fins is the high pressure loss penalty.

Vortex generators (VGs), as a passive heat transfer enhancement technique, has received widely attention for its advantage of improving the heat transfer performance with relatively low pressure loss. The mechanism of VGs is to produce secondary flows, particularly longitudinal vortices, which could induce strong swirling motion, disrupt the growth of the thermal boundary layer, bring the heat from the wall to the core of the flow, and hence lead to augmentation in heat transfer [12]. Wings or winglet type of VGs in the form of triangular and rectangular plan form are most widely investigated in the past years. These VGs can be mounted or punched out of the primary heat transfer surface with an angle of attack varying from 0° to 90°. The placement of VGs can be divided into "common flow up" and "common flow down" configurations. If the transverse distance between the trailing edge of

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Nomenclature

a	major axis of oval tube (m)
A	total heat transfer area (m^2)
b	minor axis of oval tube (m)
c_p	specific heat ($J\ kg^{-1}\ K^{-1}$)
d	lateral length of vortex generator (m)
D_c	fin collar outside diameter ($D_o + 2\delta$) (m)
D_o	hydraulic diameter ($4\pi ab/(2\pi b + 4(a - b))$) (m)
f	fraction factor
f_0	friction factor of heat exchanger without VGs
F_p	fin pitch (mm)
h	height of vortex generator (m)
h_c	convective heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
j	Colburn factor
j_0	Colburn factor of heat exchanger without VGs
l	chord length of vortex generator (m)
L	length of heat exchanger (m)
m	mass flow rate ($kg\ s^{-1}$)
Nu	Nusselt number
Δp	pressure difference between inlet and outlet of the channel (Pa)
P_l	longitudinal distance of tubes (m)
Pr	Prandtl number
P_t	transverse distance of tubes (m)
q_w	wall heat flux ($W\ m^{-2}$)
Q	heat transfer rate (W)
R	overall performance factor $(Nu/Nu_0)/(f/f_0)^{1/3}$
Re_{Dc}	Reynolds number based on fin collar outside diameter
S_1	distance between trailing edge of VGs and tube center in x direction (m)
S_2	distance between trailing edge of VGs and tube center in y direction (m)
T	temperature (K)
TG	magnitude of temperature gradient ($K\ m^{-1}$)
u_m	mean velocity at the minimum flow cross section ($m\ s^{-1}$)
VGs	vortex generators
x, y, z	Cartesian coordinates

Greek letters

β	attack angle (°)
δ_f	fin thickness (mm)
θ	synergy angle (°)
θ_s	surface averaged synergy angle (°)
λ	thermal conductivity ($W\ m^{-1}\ K^{-1}$)
v	kinematic viscosity ($m^2\ s^{-1}$)
ρ	density ($kg\ m^{-3}$)
Ω_x	dimensionless vorticity

Subscripts

s	surface averaged value
in	inlet
out	outlet
w	wall

winglet pair is shorter than that between the leading edge of the winglet pair, it is called "common flow up" configuration, as shown in Fig. 1(a). On the contrary, for the "common flow down" configuration, the transverse distance between trailing edge is longer than that between leading edge, as shown in Fig. 1(b). Torii and his coworkers [13] found that "common flow up" configuration applied in a fin and circular-tube heat exchanger could create constricted passage in the aft region of the tube, where the fluid was accelerated. As a result, the area of wake region was reduced. Since the fluid was accelerated in this region, the zone of the poor heat transfer on the surface was also removed from the near wake region. With these advantages, "common flow up" configuration was considered as the most effective heat transfer technique for low Reynolds number air-cooled flow. Joardar et al. [14] experimentally evaluated the potential of winglet type VGs by full- scale wind tunnel testing for a FTHXs. These VGs were arranged in a "common flow up" configuration with Reynolds number ranging from 220 to 960. The results showed that the heat transfer coefficient was increased 16.5–44% and the pressure drop was increased by less than 12% for the single-row winglet arrangement. He et al. [15] numerically investigated the heat transfer enhancement and pressure loss penalty for in-lined fin-tube heat exchangers with rectangular winglet pairs. The Reynolds number of the simulation ranged from 550 to 880 and the VGs were placed in a "common flow up" configuration. The parameter studies on angle of attack indicated that the heat transfer coefficient was increased by 28.8–34.5%, 54.6–61.5% and 83.3–89.7% for Deg-10 case, Deg-20 case and Deg-30 case, respectively. The corresponding pressure drop was also increased. And considering overall performance covering both heat transfer and pressure loss, Deg-10 case and Deg-20 case showed better performance.

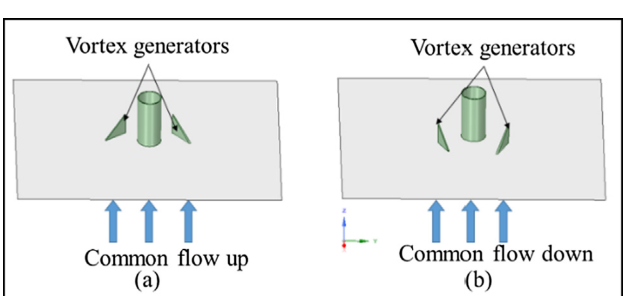

Fig. 1. Configuration of winglet type vortex generator on the fin of fin – tube heat exchangers: (a) "common flow up" configuration; (b) "common flow down" configuration.

For the purpose to optimize the heat transfer enhancement with modest pressure loss, various investigations have been carried out besides VGs layout. The combination of VGs with oval tubes instead of circular tubes was expected to achieve desirable heat transfer enhancement with low pressure loss for its streamlined configuration [16]. Chen et al. [17] numerically calculated the conjugate heat transfer of delta winglet punched from the fin surface of a fin-and-oval-tube heat exchanger. The best overall performance, $(j/j_0)/(f/f_0) = 1.04$, was obtained at $Re = 300$. Tiwari et al. [18] also performed numerical investigations on fin-and-oval-tube heat exchangers with delta winglet pairs. The results indicated that enhancement of heat transfer for the case with VGs was increased by 43.86% compared with the base case without VGs.

The VGs with a streamlined outline have emerged as a hot topic recently for the ability to reduce form drag. Zhou et al. [19] proposed a kind of VGs named curved-trapezoidal winglet and the experimental results showed that the curved-trapezoidal winglet pair has a better thermal-hydraulic performance than plane winglet pair in laminar and transitional region. Further experiments by Zhou et al. [20] indicated that punching holes directly from the surface of VGs could drastically reduce pressure loss because the