International Journal of Heat and Mass Transfer 120 (2018) 1173–1186

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Optimal design of the semi-dimple vortex generator in the fin and tube heat exchanger

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A R T I C L E I N F O

Article history:
Received 8 January 2017
Received in revised form 1 August 2017
Accepted 24 November 2017

Keywords:
Fin and tube heat exchanger
Heat transfer enhancement
CFD
Vortex generator
Semi dimple

A B S T R A C T

The fin and tube heat exchanger with a semi-dimple pair is numerically studied in terms of the air side's thermal performance. The investigated parameters consist of the diameter, attack angle and the placed location of the semi-dimple. The results show that the 30° attack angle, 5.5 mm vertical distance and 7.5 mm horizontal distance can perform the highest j/f ratio or "goodness parameter" of every semi-dimple's diameter. Additionally, the thermal performance of the fin with a semi-dimple pair is also compared with that of the fin with existing semi-dimples and the plain fin within the Reynolds number ranging from 813 to 4019. Obviously, the existing semi-dimple exhibits the highest heat transfer coefficient and pressure drop. However, the semi-dimples' pair demonstrates a goodness factor of 33–37% greater than the existing semi-dimples and 15–20% greater than the plain fin, approximately.

1. Introduction

Various industries are concerned with energy. The fin and tube heat exchanger is an appropriate device for utilizing energy. The thermal performance enhancement of the fin and tube heat exchanger affects global energy conservation, including environmental protection from the pollution that is generated from energy generation. Most thermal resistance is on the air side [1]. Therefore, the heat transfer enhancement of the air side is always selected. Passive enhancement is the most attractive method because it does not need any energy for enhancement. One of the prominent passive methods is the vortex generator (VG) attachment on the fin surface of the fin and tube heat exchanger. This method not only increases heat transfer performance but it also reduces the pressure drop in some conditions.

The first study of VG association began in 1960 by Schubauer and Spangenberg [2]. They concentrated on the impact of the VGs on boundary layer development for airplane design. In 1969, Johnson and Joubert [3] conducted the first study of the influence of VGs on thermal performance. They evaluated the heat transfer comparison of the air flowing through the normal circular tube and circular tube with delta winglet VGs, which were installed in a rectangular channel. They found that the local Nusselt number of the circular tube with VGs doubled. However, overall heat transfer coefficient did not have a significant difference. Moreover, the vortex generator can be used to attach on the fin surface of a heat sink for enhancing the heat transfer rate [4]. In 2004, Leu et al. [5] performed numerical and experimental investigations of the effect of rectangular winglets on the thermal performance and flow characteristics of the fin and tube heat exchangers under Reynolds number 400 to 3000. They concluded that the fin with VGs could increase the heat transfer rate by inducing the fluid in the wake region behind the tube. They illustrated this explanation by performing a dye injection. The fluid behind the tube flowed from the normal wake region to the next tube. This was the main cause of the increment of the heat transfer rate through adding VGs. In 2007, Wu and Tao [6] presented the effect of the attack angle of VGs on the heat transfer performance and flow characteristics through numerical evaluation. The most interesting point was the fin with VGs at the attack angle of 30°, which established a higher heat transfer rate with a lower pressure drop than a plain fin. In addition, in 2011, Wu and Tao [7] confirmed that the VG was capable of enhancing the heat transfer with pressure drop reduction when compared with a plain fin. They also presented the common flow up to expose the heat transfer performance, which reduced the pressure drop more than the common flow down. Lemouedda et al. [8] also studied the influence of the common flow up and common flow down on the heat transfer

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https://doi.org/10.1016/j.ijheatmasstransfer.2017.11.121
0017-9310/

Nomenclature

A_c	minimum flow area (m^2)
A_h	heat transfer area between the air and fin (m^2)
A_o	total sureface area (m^2)
ATA	attack angle (degree)
c_p	specific heat at constant pressure (kJ/kg·K)
D_c	fin collar outside diameter (mm)
DSD	diameter of semi dimple (mm)
f	friction factor
GF	goodness factor
$\bar{h}$	average of heat transfer coefficient ($W/m^2·K$)
HD	horizontal distance (mm)
j	Colburn factor
k	thermal conductivity (W/m·K)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (Pa)
Pr	Prandtl number
ΔP	pressure drop (Pa)
Q	heat transfer rate (W)
Re	Reynolds number
T	temperature (K)
$\bar{T}_s$	mean temperature of the surface (K)
V	velocity (m/s)
VD	vertical distance (mm)
V_{max}	maximum velocity inside the heat exchanger (m/s)
u_i	Cartesian velocity component (m/s)
Greek letters	
μ	viscosity (kg/m·s)
ρ	density (kg/m^3)
Subscripts	
a	air
in	inlet
m	average value
out	outlet

performance of the fin and tube heat exchanger. They found that the common flow up indicated higher thermal performance than the common flow down. In 2012, Wu and Tao [9] also found that the VGs that were generated by punching gave better heat transfer performance than the attached VGs because the punched hole allowed the fluid to flow across the fin channel. Many investigations of the flow and heat transfer characteristics for alternative VGs were also proposed. Zhou and Ye [10] proposed trapezoidal and curved trapezoidal winglets and compared the effects of heat transfer enhancement with rectangular and delta winglets. It was found that the trapezoidal winglets provided the highest thermohydraulic performance in laminar flow and curved trapezoidal winglets established the highest thermohydraulic performance in turbulent flow. In 2014, Zhou and Feng [11] presented curved winglets with holes for heat transfer enhancement of fin and tube heat exchangers. Normal and curved shapes of the rectangular, delta and trapezoidal winglets were investigated. They found that the punched hole on the VGs could enhance heat transfer performance.

Many VGs' geometries were investigated. They consisted of the rectangular, delta, trapezoidal and curved trapezoidal winglets, including semi-dimples. Semi-dimples were very interesting because the dimple on the fin could generate recirculation flow and horseshoe vortices with a single unit, including the hole on the fin allowing the fluid to flow across the fin to enhance heat transfer. In 2014, Wang et al. [12,13] conducted experimental investigations of the flow and heat transfer characteristics of fin and tube heat exchangers with the semi-dimple VGs and compared the heat transfer performance with the heat transfer performances of a plain fin and a louver fin within frontal velocity ranging from 0.5 to 5 m/s. One tube row and two tube rows with 3 mm diameter semi-dimples and 2 fin pitches, which were 1.6 mm and 2.0 mm, were investigated. The result was that the louver fin produces the highest heat transfer coefficient and pressure drop in almost all conditions. But fins with semi-dimples conducted a higher heat transfer coefficient than the louver fin within a range of frontal velocity lower than 2 m/s for 2 mm of the fin pitch.

Although the pressure drop obtained from the fins with semi-dimples was lower than louver fins, it was still large when compared with that of the plain fin. Because of 5 semi-dimples on the fin block, the fluid flowed through the heat exchanger, so the pressure drop increased. Furthermore, the semi- dimple's attack angle of 90° could provide the maximum frontal area which resulted in the obstruction of fluid flow in the channel. The semi-dimple's attack angle of 90° fully impeded the fluid flow. The aim of this study is to optimize the heat transfer performance of the fin with a pair of semi-dimples by investigating the effects of attack angle, semi-dimple position and semi-dimple size on the flow and heat transfer characteristics. The numerical solution method is used for this study. The effect of attack angle, semi-dimple position and semi-dimple size on the flow and heat transfer characteristics are reported. Moreover, the heat transfer performances of the optimal design of the fins with semi-dimple are also compared with the plain fin and the fins with existing semi-dimples.

2. Mathematical analysis

2.1. Computational models

Fig. 1 shows the schematic view of the fin and the semi-dimple in this study. The fin geometry in Fig. 1(a) is only a single channel of the fin with fin pitch of 1.6 mm, fin thickness of 0.11 mm, collar diameter of 7.5 mm, transverse pitch of 21 mm and longitudinal pitch of 18.2 mm [13], including the semi-dimple with a diameter of 1–2.6 mm. Moreover, each semi-dimple pair that is punched on the fin is placed on different locations with different attack angles. The details of the position, attack angle and diameter of the semi-dimple are shown in Table 1.

The horizontal and vertical positions are measured from the center of the tube to the center of the semi-dimple as shown in Fig. 1(b). The attack angle is the angle between the flow direction and cut line of the semi-dimple as shown in Fig. 1(c). In the case of the computational domain, a half fin with a semi-dimple is determined as shown in Fig. 2. The entrance length is 43 mm and the exit length is 151 mm as shown in Fig. 2(a). The exit length needs to be longer than 7 times the longitudinal pitch to prevent the reverse flow. Therefore, 135 different configurations of fin with semi-dimples are evaluated via numerical method by the commercial software "ANSYS CFX 17.0."

2.2. Governing equations and parameter definitions

The governing equations on behalf of three dimensions of the Cartesian coordinate, which are steady, incompressible and constant thermal property, are identified below