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Evaporation heat transfer and pressure drop in flattened microfin tubes having different aspect ratios

E.J. Lee, N.H. Kim *

Department of Mechanical Engineering, University of Incheon, 12-1, Songdo-Dong, Yeonsu-Gu, Incheon 406-772, Republic of Korea

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ABSTRACT

In this study, evaporation heat transfer coefficients and pressure drops of R-410A were obtained in flattened microfin tubes (AR = 2, 4) made from 7.0 mm O.D. round microfin tubes. The test range covered mass flux 200–400 kg/m^2 s, heat flux from 5 to 15 kW/m^2 and saturation temperature from 10 to 15 °C. The evaporation heat transfer coefficient increases as mass flux or heat flux increases. The heat transfer coefficient also increases as aspect ratio increases. The frictional pressure drop increases as quality or mass flux increases, saturation temperature decreases, and is independent of heat flux. The frictional pressure drop also increases as aspect ratio increases. Comparison with existing round microfin tube correlations is made.

1. Introduction

A special enhanced copper round tube commonly called the microfin tube is widely used for fin-and-tube evaporators and condensers of residential air conditioners or heat pumps. Typical round microfin tubes have an outside diameter from 4 to 15 mm, 50 to 70 fins with helix angle (β) from 6° to 30°, fin height from 0.1 to 0.25 mm, fin apex angle (γ) from 25° to 70° [1–3]. It is known that microfins significantly enhance the heat transfer with marginal pressure drop increase. For evaporation, heat transfer enhancement is realized by increase of heat transfer area and turbulence generated by the fins. Early transition from wavy-stratified flow to annular flow is also responsible for the heat transfer enhancement [4].

Round tubes of fin-and-tube heat exchangers, however, inevitably induce low thermal performance regions downstream of the tubes. Usage of oval or flat tubes instead of round tubes will mitigate the air-side performance degradation. The amount of refrigerant charge will also be reduced compared with that in the round tube [5]. Webb and Iyengar [6] compared the air-side performance of the fin-and-tube heat exchanger having oval tubes (5 mm × 8 mm) with that of the fin-and-tube heat exchanger having round tubes (O.D. = 8 mm). The heat transfer coefficient of the oval tube heat exchanger was approximately the same as that of the round tube heat exchanger. The pressure drop of the oval tube heat exchanger, however, was 10% lower. Similar observation was reported by Kim and Kim [7] from the air-side performance comparison of the fin-and-tube heat exchanger with flat tubes (3.5 mm × 9.5 mm) and the fin-and-tube heat exchanger with round tubes (O.D. = 7.0 mm).

Literature reveals many studies on evaporation in round tubes [1,2,8,9]. However, investigations on evaporation in oval or flat tubes are very limited. Kim et al. [10] obtained the R-22 evaporation heat transfer coefficient in an oval microfin tube of 1.5 aspect ratio, which was made by deforming the 9.5 mm O.D. microfin tube. The microfin tube had 60 fins of 0.2 mm fin height with 18° helix angle. The mass flux was varied from 150 to 300 kg/m^2 s at fixed heat flux of 12 kW/m^2. The heat transfer coefficient of the oval tube was 2–15% higher than that of the round tube. The pressure drops were approximately the same. Moreno Quiben et al. [11,12] obtained R-22 and R-410A evaporation heat transfer coefficients and the pressure drops in smooth flat tubes having 2 mm or 3 mm internal height. The flat tubes were made from 8.0 mm I.D and 13.8 mm I.D. round tubes respectively. The mass flux was varied from 150 to 500 kg/m^2 s, and the heat flux was varied from 6 to 40 kW/m^2. Both heat transfer coefficients and pressure drops of the 2 mm height tube were higher than those of the 3 mm height tube. Comparison with existing correlations revealed that evaporation heat transfer coefficients were predicted reasonably well with usage of the equivalent diameter. Pressure drops were, however, highly underpredicted. Nasr et al.

* Corresponding author. Tel.: +82 32 835 8420; fax: +82 32 835 8410.
E-mail addresses: hvac@incheon.ac.kr (E.J. Lee), knh0001@incheon.ac.kr (N.H. Kim).

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Nomenclature

A	area, m^2
AR	aspect ratio (=h/w)
Co	confinement number
c_p	specific heat, J/kg K
D	diameter, m
D_e	equivalent diameter, m
D_h	hydraulic diameter, m
D_m	melt-down diameter, m
D_r	fin root diameter, m
D_t	fin tip diameter, m
e	fin height, m
F	modified Froude number
G	mass flux, kg/m^2 s
g	gravitational constant, m/s^2
h	heat transfer coefficient, W/m^2 K or tube height, m
i_{fg}	latent heat of vaporization, J/kg
k	thermal conductivity, W/m K
$\dot{m}$	mass flow rate, kg/s
Nu_{Dh}	Nusselt number based on hydraulic diameter
Pr	liquid Prandtl number
P_w	wetted perimeter, m
Q	heat transfer rate, W
q	heat flux, W/m^2
Re_{Dh}	Reynolds number based on hydraulic diameter
t	tube wall thickness, m
T	temperature, K
U	overall heat transfer coefficient, W/m^2 K
w	tube width, m
x	quality
X_{tt}	Martinelli parameter
dp/dz	pressure gradient, Pa/m
α	void fraction
β	fin helix angle, degree
γ	fin apex angle, degree
ρ	density, kg/m^3
σ	surface tension, N/m

Subscripts

ave	average
c	cross-sectional
exp	experimental
f	friction
g	gas
i	inside
in	inlet
l	liquid
m	middle or melt-down
o	outside
p	preheater
pred	prediction
r	refrigerant or fin root
sat	saturation
sens	sensible
t	fin tip
w	water or tube wall

[13] measured R-410A evaporation heat transfer coefficient in smooth flat tubes made from 8.7 mm I.D. tube. Four tubes with internal height 2.8, 3.8, 5.5 and 6.6 mm were tested. The investigated range of mass flux (74–107 kg/m^2 s) and the heat flux (1.5–4.0 kW/m^2) was rather low. Both heat transfer coefficient and pressure drop increased with the aspect ratio. Kim et al. [14] obtained R-410A evaporation heat transfer coefficients and pressure drops in flat smooth tubes. The test range covered mass flux 200–400 kg/m^2 s, heat flux from 5 to 15 kW/m^2. Evaporation heat transfer coefficients and pressure drops increased with the increase of aspect ratio.

The foregoing literature survey reveals that investigations on the evaporation in oval or flat tube are very limited. Especially for evaporation heat transfer in flat microfin tube, the study by Kim et al. [10] is the only one available. The primary microfin tube used by them had 9.5 mm O.D. Recent trend of fin and tube heat exchanger is to use smaller diameter tubes. Usage of smaller diameter tubes yields higher air-side heat transfer coefficient and lower pressure drop due to reduced wake region behind the tubes [1]. In this study, 7.0 mm O.D. microfin tube was progressively deformed to yield flat tubes having two different aspect ratios 2 and 4 (internal heights to the fin root 4.08 and 2.25 mm respectively). Evaporation heat transfer and pressure drop data were obtained using R-410A. Mass flux was varied from 200 to 400 kg/m^2 s, heat flux was varied from 5 to 15 kW/m^2 and saturation temperature were varied from 10 to 15 °C. A smooth tube having the same O.D. (7.0 mm) with the microfin tube was also tested.

2. Experimental apparatus and procedures

2.1. Flat tube samples

Commercial microfin tube used in this study is shown in Fig. 1. The 7.0 mm O.D. microfin tube has 65 microfins of 0.1 mm height with 15° helix angle. Geometric details of the microfin tube are provided in Table 1. Round microfin tube was flattened using special dies of predetermined aspect ratio (AR) of 2 and 4. Cross-sectional photos of resultant flat tubes are shown in Fig. 2. The saturation pressure of R-410A corresponding to the room temperature 30 °C is 1.89 MPa. At the high pressure, bare flat tubes will deform. To prevent deformation at the high pressure, annular copper bars of 2.0 mm thickness, which exactly fitted flat tubes, were soldered at outside of the tubes. Before soldering, both surfaces of copper bar and flat tube were mildly scratched with coarse sand paper for smooth flow of solder. Perfect soldering was checked by cutting and inspecting the cross section at several places along longitudinal direction. A schematic drawing of the test tube is shown in Fig. 4(c). Geometric details were measured from enlarged photos of the cross-section, and are listed in Table 1. To confirm the possible deformation at high internal pressure, flat tubes were pressurized to 3.0 MPa for three days. No measurable deformation was noticed.

2.2. Experimental apparatus

Detailed explanation on the apparatus is provided in Kim et al. [14], and only short summary will be provided. As illustrated in Fig. 3, the refrigerant flows into the test section at a known quality and evaporates in the test section by hot water flowing in the annulus. Two-phase refrigerant mixture out of the test section fully condenses in the shell-and-tube heat exchanger located at downstream of the test section by a circulating brine. The condensed refrigerant passes through the magnetic pump, mass flow meter, and is supplied to the pre-heater. The refrigerant flow rate is controlled by by-passing an appropriate amount of liquid from the pump. The vapor quality into the test section is controlled by the heat input supplied to the pre-heater. The heat flux to the flat tube is controlled by changing the temperature of hot water in the