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

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ABSTRACT

In this study, condensation heat transfer coefficients and pressure drops of R-410A are obtained in flattened microfin tubes made from 7.0 mm O.D. round microfin tubes. The test range covers saturation temperature 45 °C, mass flux 100–400 kg m^{-2} s^{-1} and quality 0.2–0.8. Results show that the effect of aspect ratio on condensation heat transfer coefficient is dependent on the flow pattern. For annular flow, the heat transfer coefficient increases as aspect ratio increases. For stratified flow, however, the heat transfer coefficient decreases as aspect ratio increases. The pressure drop always increases as aspect ratio increases. Possible reasoning is provided based on the estimated flow pattern in flat microfin tubes. Comparison with existing round microfin tube correlations is made.

Transfert de chaleur de condensation et chute de pression dans des tubes à micro-ailettes aplatie avec différents rapports longueur-diamètre

Mots clés : Condensation ; Coefficient de transfert de chaleur ; Chute de pression ; Micro ailettes ; Tube ; R410A

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Nomenclature

A	Area, m^2
AR	Aspect ratio ($=h/w$)
c_p	Specific heat, $J\ kg^{-1}\ K^{-1}$
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
G	Mass flux, $kg\ m^{-2}\ s^{-1}$
g	Gravitational constant, $m\ s^{-2}$
h	Heat transfer coefficient, $W\ m^{-2}\ K^{-1}$ or tube height, m^{-1}
i_{fg}	Latent heat of vaporization, $J\ kg^{-1}$
J_g	Regime criteria defined by $xG/[gD\rho_v(\rho_l-\rho_v)]^{0.5}$
k	Thermal conductivity, $W\ m^{-1}\ K^{-1}$
$\dot{m}$	Mass flow rate, $kg\ s^{-1}$
Nu_{Dh}	Nusselt number based on hydraulic diameter
Pr	Liquid Prandtl number
P_w	Wetted perimeter, m
Q	Heat transfer rate, W
Re_{Dh}	Reynolds number based on hydraulic diameter
t	Tube wall thickness, m
T	Temperature, K
U	Overall heat transfer coefficient, $W\ m^{-2}\ K^{-1}$
w	Tube width, m
x	Quality
X_{tt}	Martinelli parameter
dp/dz	Pressure gradient, $Pa\ m^{-1}$
α	Void fraction
β	Fin helix angle, degree
γ	Fin apex angle, degree
ρ	Density, $kg\ m^{-3}$

Subscripts

ave	Average
c	Cross-sectional
exp	Experimental
f	Friction
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
v	Vapor
w	Water or tube wall

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° (Webb and Kim, 2005; Cavallini et al., 2009; Laohalertdecha et al., 2012). It is known that microfins significantly enhance the heat transfer with marginal pressure drop increase. For condensation, heat transfer enhancement is realized by increase of heat transfer area, turbulence and surface tension induced drainage by the fins. Early transition from wavy-stratified flow to annular flow is also responsible for the heat transfer enhancement (Cavallini et al., 2000).

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 air-side performance degradation. The amount of refrigerant charge will also be reduced compared with that in the round tube (Wilson et al., 2003). Webb and Iyengar (2001) 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 (2010) 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 condensation or evaporation in round tubes (Cavallini et al., 2009; Webb and Kim, 2005; Collier and Thome, 1994; Ghiaansiaan, 2008). However, investigations on condensation or evaporation in oval or flat tubes are very limited. Wilson et al. (2003) measured R-22 and R-410A condensation heat transfer coefficients and pressure drops in flat tubes, which were made by gradually deforming the 9.5 mm O.D. round smooth or microfin tube. The microfin tube had 60 fins of 0.2 mm fin height with 18° helix angle. The mass flux was varied from 75 to 400 $kg\ m^{-2}\ s^{-1}$. Condensation heat transfer coefficient increased with the aspect ratio of the tube. Maximum heat transfer coefficient was obtained for the tube having 2.57 mm internal height. The maximum enhancement ratio over round smooth tube was approximately twofold for the smooth flat tube, and fivefold for the microfin flat tube. Kim et al. (2001) 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 $kg\ m^{-2}\ s^{-1}$ at fixed heat flux of 12 $kW\ m^{-2}$. The heat transfer coefficient of the oval tube was 2–15% higher than