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Condensation of pure and near-azeotropic refrigerants in microfin tubes: A new computational procedure

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ABSTRACT

Microfin tubes are widely used in air cooled and water cooled heat exchangers for heat pump and refrigeration applications during condensation or evaporation of refrigerants. In order to design heat exchangers and to optimize heat transfer surfaces, accurate procedures for computing pressure drops and heat transfer coefficients are necessary. This paper presents a new simple model for the prediction of the heat transfer coefficient to be applied to condensation in horizontal microfin tubes of halogenated and natural refrigerants, pure fluids or nearly azeotropic mixtures. The updated model accounts for refrigerant physical properties, two-phase flow patterns in microfin tubes and geometrical characteristics of the tubes. It is validated against a data bank of 3115 experimental heat transfer coefficients measured in different independent laboratories all over the world including diverse inside tube geometries and different condensing refrigerants among which R22, R134a, R123, R410A and CO_2.

Condensation des frigorigènes purs et quasi-azéotropiques à l’intèrieur de tubes à microailettes : une nouvelle procédure numérique

Mots clés : Échangeur de chaleur ; Tube microaileté ; Calcul ; Transfert de chaleur ; Chute de pression ; Condensation ; Frigorigène ; Dioxyde de carbone

1. Introduction

In order to reduce both the refrigerant charge and the volume of machinery, the refrigeration and air conditioning manufactories design their heat exchangers with enhanced surfaces. Microfin tubes are widely used for condensation and vaporization in heat pumps, chillers and HVAC systems.

Typical microfin tubes available for industrial applications are made of copper and have an outside diameter from 4 to 15 mm, a single set of 50–70 spiral fins with spiral angle (β)

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Nomenclature

$Bo = g\rho_L h\pi D/(8\sigma n_g)$ (Bond number)
c_p specific heat capacity [$J\,kg^{-1}\,K^{-1}$]
D fin tip diameter [m]
e (percent deviation) = $[(\alpha_{CALC} - \alpha_{EXP})/\alpha_{EXP}]100$
e_{AB} (mean absolute deviation) = $(1/Np)\Sigma[|e|]$
e_R (average deviation) = $(1/Np)\Sigma[e]$
$Fr = G^2/[(\rho_L - \rho_G)^2 gD]$ (Froude number)
$Fr_{GO} = G^2/(\rho_G^2 gD)$
g gravitational acceleration [$m\,s^{-2}$]
$G = 4\,m/(\pi D^2)$ mass velocity [$kg\,m^{-2}\,s^{-1}$]
h fin height [m]
h_{LG} latent heat [$J\,kg^{-1}$]
$J_G = xG/[gD\rho_G(\rho_L - \rho_G)]^{0.5}$
L tube length [m]
m mass flow rate [$kg\,s^{-1}$]
Np number of experimental data points
n_g number of grooves
$Pr = \mu c_p/\lambda$ (Prandtl number)
$Pr_L = \mu_L c_{pL}/\lambda_L$
p_{RED} reduced pressure
q heat flow rate [W]
$Re = GD/\mu$ (Reynolds number)
$Re_{eq} = Re_{LO}\,[(1-x) + x\,(\rho_L/\rho_G)^{1/2}]$
$Re_{LO} = GD/\mu_L$ all liquid Reynolds number
Rx geometry enhancement factor (Eq. (10))
T temperature [K]
$\Delta T = (T_S - T_W)$ [K]
T_S saturation temperature [K]
T_W tube internal wall temperature [K]
x thermodynamic vapour quality
$X_{tt} = (\mu_L/\mu_G)^{0.1}\,(\rho_G/\rho_L)^{0.5}\,[(1-x)/x]^{0.9}$

Greek symbols
α heat transfer coefficient [$W\,m^{-2}\,K^{-1}$]
β spiral angle [rad]
γ apex angle [rad]
λ thermal conductivity [$W\,m^{-1}\,K^{-1}$]
μ dynamic viscosity [$kg\,m^{-1}\,s^{-1}$]
ρ density [$kg\,m^{-3}$]
σ_N (standard deviation) = $\{\,[\,\Sigma\,(e - e_R)^2\,]/(Np - 1)\,\}^{1/2}$

Subscripts
A ΔT-independent flow regime
CALC calculated
D ΔT-dependent flow regime
EXP experimental
G gas phase
GO gas phase with total flow rate
L liquid phase
LO liquid phase with total flow rate

from 6 to 30°, fin height (h) from 0.1 to 0.25 mm, fin apex angle (γ) from 25 to 90°. Fig. 1 presents the characteristic geometrical parameters of microfin enhanced tubes.

In condensation, microfin tubes show a heat transfer enhancement when compared to equivalent plain tubes at the same operating conditions (Cavallini et al., 2002a,b, 2006a); on the other hand, these tubes also present higher pressure losses than the plain tubes.

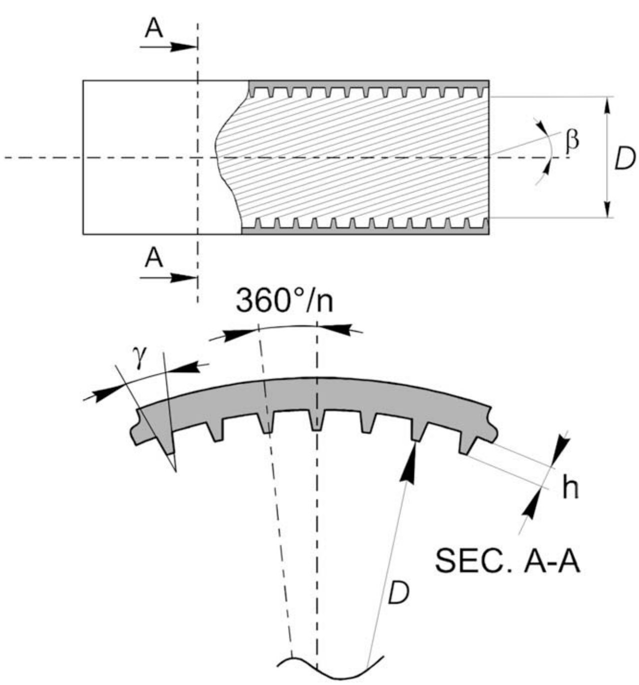

Fig. 1 – Characteristic geometrical parameters of microfin tubes.

The same authors (Cavallini et al., 2000) also pointed out that the heat transfer enhancement could be explained by three different reasons: first of all the mere increase of the effective heat exchange area. Additionally, the turbulence induced in the liquid film and the surface tension effect on the liquid drainage promote an early transition from the wavy-stratified flow to the annular flow.

Doretti et al. (2005) and Cavallini et al. (2006b) have experimentally visualized the two-phase flow pattern during condensation of halogenated refrigerants inside a microfin tube aiming to link the heat transfer enhancement to the flow pattern transition.

Several authors have experimentally measured the heat transfer coefficient of halogenated and natural refrigerants, pure fluids or nearly azeotropic mixtures, in enhanced tubes. Present authors have collected around 3100 experimental measures of mean, quasi-local and local condensation heat transfer coefficients in order to critically review the models available in the open literature and to develop a new computational procedure for heat transfer and pressure drop prediction. Table 1 summarizes all the experimental test runs in this data bank.

In recent years, many researchers such as Cavallini et al. (1999), Chamra et al. (2005), Chamra and Mago (2006), Han and Lee (2005), Kedzierski and Goncalves (1997), Koyama and Yonemoto (2006), Yu and Koyama (1998) and Wang et al. (2007) have suggested different simple semi-empirical models for the calculation of the heat transfer coefficient during condensation in horizontal helical microfin tubes.