Two-phase heat transfer characteristics of R410A-oil mixture flow condensation inside small diameter microfin copper tubes

Guoliang Ding [a*], Haitao Hu [a], Xiangchao Huang [a], Yu Zhu [a], Yifeng Gao [b], Yongxin Zheng [b], Ji Song [b]

([a] Institute of Refrigeration and Cryogenics, Shanghai Jiaotong University, Shanghai 200240, China)

([b] International Copper Association Shanghai Office, Shanghai 200020, China)

* Corresponding Author, Tel: 86-21-34206378; Fax: 86-21-34206814; E-mail: glding@sjtu.edu.cn

Abstract

For the application of small diameter copper tube, two-phase heat transfer characteristics of R410A-oil mixture flow condensation inside small diameter microfin copper tubes were investigated experimentally. The test results indicate that the presence of oil deteriorates the heat transfer. The deterioration effect is negligible at nominal oil concentration of 1%, and becomes obvious with the increase of nominal oil concentration. At 5% nominal oil concentration, the heat transfer coefficient of R410A-oil mixture is found to have a maximum reduction of 25.1% for small diameter microfin tubes. By substituting the mixture's properties, Yu and Koyama correlation is recommended to predict the local condensation heat transfer coefficient of R410A-oil mixture inside small diameter microfin tubes.

Keywords: Condensation; Correlation; Heat transfer; Oil; R410A; Small diameter copper tube

Nomenclature

A	heat transfer surface area (m^2)	ε	surface area enhancement
c_p	isobaric specific heat ($J\ kg^{-1}\ K^{-1}$)	λ	thermal conductivity ($W\ m^{-1}\ K^{-1}$)
d	diameter (m)	μ	dynamic viscosity (Pa s)
Fr	Froude number, dimensionless	ρ	density ($kg\ m^{-3}$)
G	mass flux ($kg\ m^{-2}\ s^{-1}$)	ω	oil concentration
Ga	Galileo number, dimensionless	Φ	two-phase frictional multiplier
h_{fg}	latent heat of condensation ($kJ\ kg^{-1}$)	*Subscripts*	
m	mass flow rate ($kg\ s^{-1}$)	bub	bubble
Nu	Nusselt number, dimensionless	e	equivalent
Ph	phase change number, dimensionless	L	liquid
Pr	Prandtl number, dimensionless	local	local
q	heat flux ($W\ m^{-2}$)	no	nominal
Re	Reynolds number, dimensionless	o	oil
T	temperature (^oC)	r	refrigerant
x	vapor quality	sat	saturated
X_{tt}	Martinelli parameter, dimensionless	tp	two-phase
Greek symbols		V	vapor
α	heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)	wi	inside tube wall
δ	void fraction		

1. Introduction

During recent years, the utilization of copper in HVAC industry is challenged by aluminum due to the high price ratio of copper to aluminum. In reality, the performance of aluminum round tube based coil versus the original copper tube based coil drops by up to 10%. Moreover, aluminum fin and tube combinations must be carefully matched and tested to provide sufficient corrosion protection, or else the durability of the fins will be affected. In an all-aluminum design, the fins are corroded faster than the tube. This means that the performance of all aluminum design degrades faster than all copper or Copper Tube Aluminum Fin (CTAF) designs.

According to above consideration, it still makes good economic sense for HVAC manufacturers to use small diameter microfin copper tube in heat exchangers compared with currently available aluminum alternatives, because small diameter microfin tube heat exchangers are cost competitive in terms of raw material and require little new tooling investment. Furthermore, by using small diameter microfin copper tube, HVAC manufacturers can maintain product performance and reliability, apply copper's antimicrobial benefits, and decrease the carbon footprint of their HVAC products. Therefore, using small diameter microfin copper tube is an attractive choice for HVAC manufacturers.

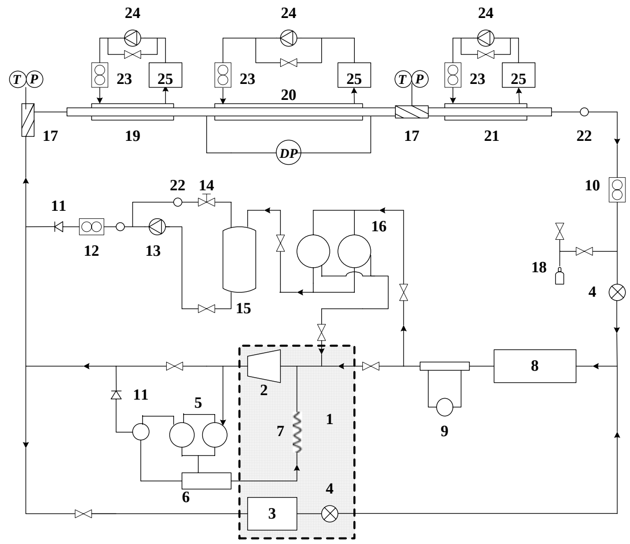
In the room air conditioners using small diameter copper tube, R410A is the mainly employed refrigerant; meanwhile, a certain amount of oil is needed for lubricating and sealing the compressor, and circulates with the refrigerant inevitably, meaning the working fluids flowing inside the tubes are refrigerant-oil mixture. For the sake of achieving good designs for room air conditioners using small diameter microfin copper tubes, heat transfer characteristics of refrigerant-oil mixtures flow condensation inside these tubes should be investigated.

Heat transfer characteristics of pure R410A (containing no oil) flow condensation inside microfin tubes have been reported by many researchers, such as Bogart and Thors [1], Cavallini et al. [2], Dunn [3], Eckels and Tesene [4], Goto et al. [5], Han and Lee [6], Jung et al. [7], Kedzierski and Goncalves [8], Kim and Shin [9], Kwon et al. [10], Miyara et al. [11], and Tang et al. [12]. However, there is no research on the heat transfer characteristics of R410A-oil mixture flow condensation inside small diameter tubes. The researches on condensation heat transfer characteristics of other refrigerant-oil mixtures inside microfin tubes show that flow condensation heat transfer characteristics of refrigerant-oil mixtures are affected by tube diameters and the properties of refrigerant-oil mixtures.

The purpose of this study is to investigate the flow condensation heat transfer characteristics of R410A-oil mixture inside small diameter microfin copper tubes of 5 mm and 4 mm O.D., and to propose a correlation for predicting the local heat transfer coefficient of R410A-oil mixture flow condensation inside small diameter microfin tubes.

2. Experimental apparatus and test conditions

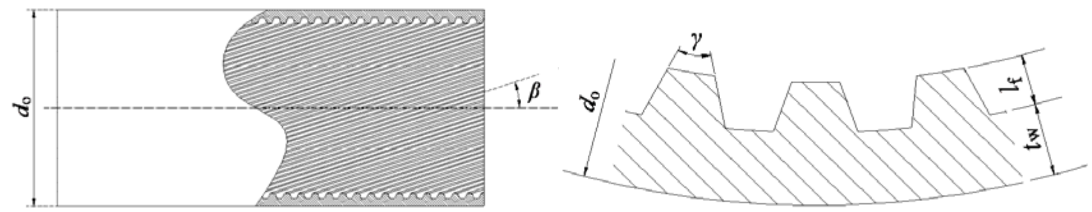
The experimental rig consists of a refrigerant loop, a lubricating oil loop and a cooling water loop, as shown in Fig. 1. The refrigerant loop is designed to measure the heat transfer characteristics of refrigerant-oil mixture; the lubricating oil loop is designed to provide the required oil concentration of test section for investigating the influence of oil on condensation heat transfer and pressure drop performance; the cooling water loop is designed to achieve the condensation of refrigerant.



1-Outdoor unit, 2-Compressor, 3-Condenser, 4-Electronic expansion valve, 5-Oil separator for compressor, 6-Oil container, 7-Capilary, 8-Indoor unit, 9-After-heater, 10-Refrigerant mass flowmeter, 11-Check valve, 12-Oil mass flowmeter, 13-Oil pump, 14-Regulating valve, 15-Oil tank, 16-Oil separator for test section, 17-Mixing chamber, 18-Sampling cylinder, 19-Pre-condenser, 20-Test section, 21-After-condenser, 22-Sight glass, 23-Water volume flowmeter, 24-Water pump, 25-Thermostat

Fig. 1. Schematic diagram of experimental rig

The test tubes used in this study are small diameter horizontal microfin copper tubes with outside diameters of 5 mm and 4 mm, respectively. The geometries of microfin tubes are shown in Fig. 2. The test fluids are R410A and ester oil RB68EP. The ester oil has good miscibility with R410A and it is one of the most common ester oils for R410A air conditioner. The thermodynamic properties of oil and R410A-oil mixture can be seen in Hu et al. (2008) [13].



(a) Cutaway view of the test tube

(b) Cross section of fin

5 mm O.D. microfin tube:	**4 mm O.D. microfin tube:**
$\beta = 18°$;	$\beta = 12°$;
$\gamma = 40°$;	$\gamma = 40°$;
$t_w = 0.20$ mm;	$t_w = 0.22$ mm;
$l_f = 0.14$ mm;	$l_f = 0.12$ mm;
$n_f = 40$.	$n_f = 40$.

Fig. 2. The geometries of small diameter microfin copper tubes

The test conditions are tabulated in Table 1. The maximum uncertainty of heat transfer coefficient is ±12.5% at typical test conditions without oil, which is estimated based on the analysis of error propagation reported by Moffat [14].

Table 2. Test conditions

Tube diameter, mm	Mass flux, kg m^{-2} s^{-1}	Heat flux, kW m^{-2}	Inlet quality	Condensing temperature, °C	Oil concentration, wt. %
5 (O.D.)	200	4.21	0.3 ~ 0.9	40	0, 1, 3, 5
	300	6.32			
	400	8.42			
4 (O.D.)	400	6.33			
	500	7.91			

3. Data reduction and uncertainty

3.1 Heat transfer coefficient

The local flow condensation heat transfer coefficients for pure refrigerants and refrigerant-oil mixtures are defined by the following equations [15]:

$$\alpha_{\mathrm{tp,r}} = q/\left(T_{\mathrm{sat}} - T_{\mathrm{wi}}\right) \tag{1}$$

$$\alpha_{\mathrm{tp,r,o}} = q/\left(T_{\mathrm{bub}} - T_{\mathrm{wi}}\right) \tag{2}$$

where

$\alpha_{\mathrm{tp,r}}$ – heat transfer coefficient for two-phase pure refrigerant, W m^{-2} K^{-1}
$\alpha_{\mathrm{tp,r,o}}$ – heat transfer coefficient for two-phase refrigerant-oil mixture, W m^{-2} K^{-1}
q – heat flux, W m^{-2}
T_{wi} – inside tube wall temperature, °C
T_{sat} – saturation temperature of the refrigerant, °C
T_{bub} – bubble point temperature of the refrigerant-oil mixture, °C

The saturation temperature of the refrigerant, T_{sat}, is determined by the vapor pressure curve of the pure refrigerant and the measured pressure; while the bubble point temperature of the refrigerant-oil mixture, T_{bub}, is determined by using the vapor-liquid equilibrium prediction method [15].

3.2 Oil concentration

Nominal oil concentration and local oil concentration are defined by Eqs. (3) and (4), respectively [15]:

$$\omega_{\mathrm{no}} = m_{\mathrm{o}}/\left(m_{\mathrm{o}} + m_{\mathrm{r}}\right) \tag{3}$$

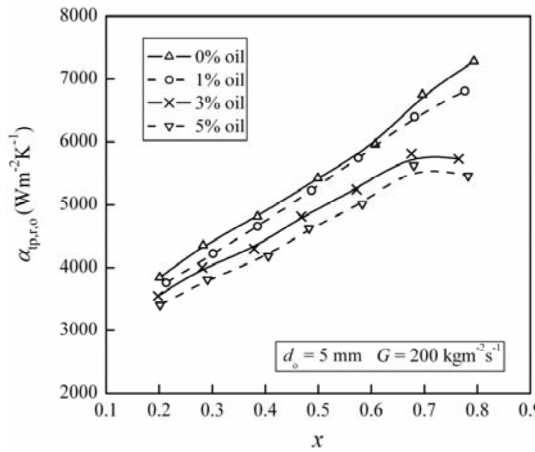
$$\omega_{\mathrm{local}} = \frac{m_{\mathrm{o}}}{m_{\mathrm{o}} + m_{\mathrm{r,L}}} = \frac{\omega_{\mathrm{no}}}{1 - x_{\mathrm{r,o}}} \tag{4}$$

where

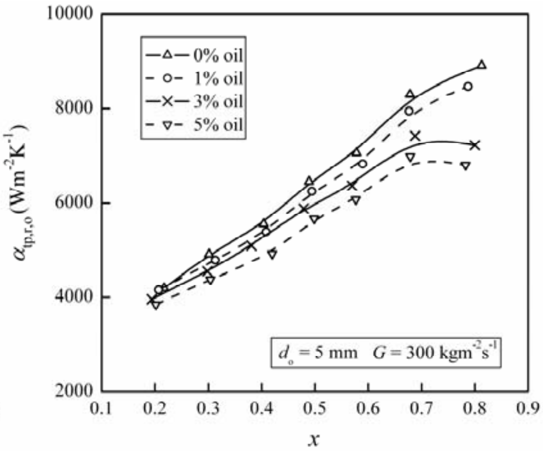
ω_{no} – oil mass fraction in subcooled liquid before evaporation begins
ω_{local} – oil mass fraction in liquid phase of refrigerant-oil mixture
$x_{\mathrm{r,o}}$ – local vapor quality of refrigerant-oil

4 Experimental results and analysis

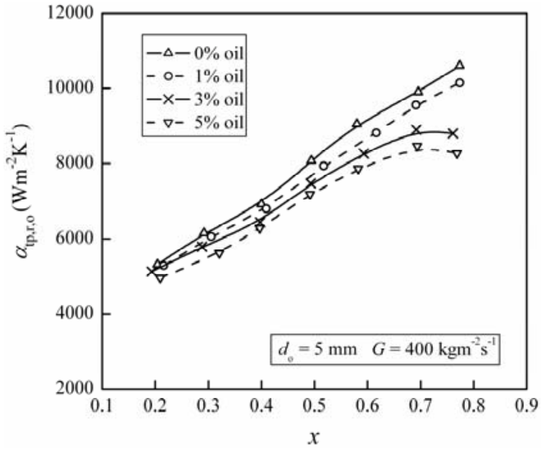
Figure 3 shows the local heat transfer coefficients of R410A-oil mixture flow condensation inside 5 mm and 4 mm O.D. microfin tubes as a function of vapor quality at different mass fluxes.



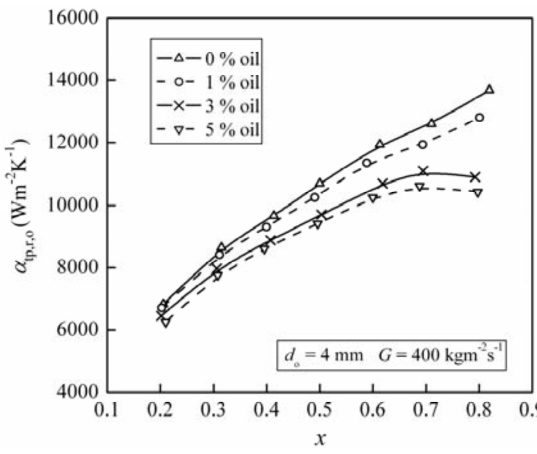
(a) $d_o = 5$ mm, $G = 200$ kg m^{-2} s^{-1}



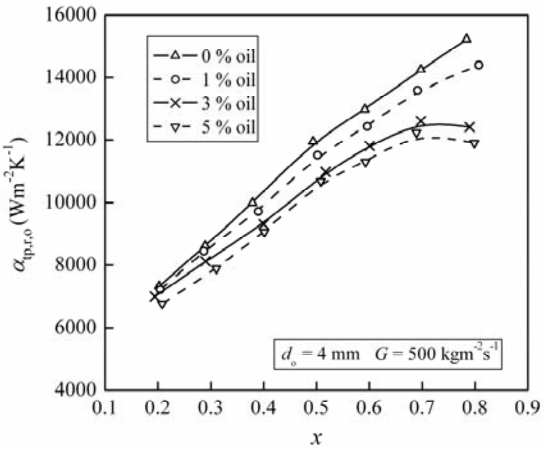
(b) $d_o = 5$ mm, $G = 300$ kg m^{-2} s^{-1}



(c) $d_o = 5$ mm, $G = 400$ kg m^{-2} s^{-1}



(d) $d_o = 4$ mm, $G = 400$ kg m^{-2} s^{-1}



(e) $d_o = 4$ mm, $G = 500$ kg m^{-2} s^{-1}

Fig. 3. Heat transfer coefficients of R410A-oil mixture flow condensation inside 5 mm and 4 mm O.D. microfin tubes as a function of vapor quality at different mass fluxes

For pure R410A and R410A-oil mixture at 1% nominal oil concentration, the condensation heat transfer coefficient decreases with the decrease of vapor quality. However, for R410A-oil mixture at 3% and 5% nominal oil concentrations, with the decrease of vapor quality, the condensation heat transfer coefficient initially increases and then decreases, presenting a peak of condensation heat transfer coefficient between the vapor quality of 0.7 and 0.75.

The reason of the decrease of condensation heat transfer coefficient with the decrease of vapor quality for pure R410A and R410A-oil mixture at 1% nominal oil concentration is that, the decrease of vapor quality may increase the heat transfer resistance, resulted from thicker condensing liquid film. The occurrence of a peak of condensing heat transfer coefficients for R410A-oil mixture at 3% and 5% nominal oil concentrations attributes to two opposite effects of vapor quality decrease on heat transfer: 1) negative effect of vapor quality decrease by increasing the refrigerant-oil mixture liquid film thickness; 2) positive effect of vapor quality decrease by decreasing the viscosity and mass transfer resistance effect in refrigerant-oil mixture liquid film. At a vapor quality of higher than the peak corresponding vapor quality, the positive effect dominates; at a vapor quality of lower than the peak corresponding vapor quality, the negative effect dominates; and at the peak corresponding vapor quality, these two effects reach equilibrium.

The peak phenomenon of condensation heat transfer coefficient at 3% and 5% nominal oil concentrations is not observed at 1% nominal oil concentration, and the possible reason is that the influence of oil on viscosity and mass transfer resistance effect in the liquid film is not apparent at small nominal oil concentration, causing an insignificant effect of oil on heat transfer coefficient.

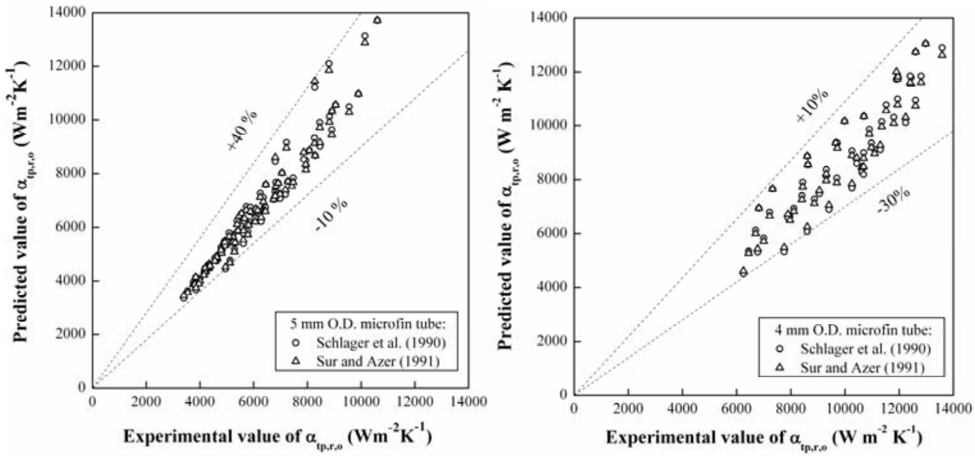
5 Heat transfer coefficient correlation for R410A-oil mixture flow condensation in microfin tubes

Many researchers have proposed their methods to predict heat transfer coefficients of refrigerant-oil mixture flow condensation in microfin tubes. Although these methods can provide satisfactory predictions to their own experimental data, none of them has a general applicability to all the refrigerant-oil mixtures. Therefore, the predictabilities of these prediction methods to the experimental data of R410A-oil mixture in small diameter microfin copper tubes are uncertain and should be investigated. For the utilization of small diameter copper tube in HVAC industry, it is necessary to find a suitable heat transfer correlation to predict the condensation heat transfer characteristic of R410A-oil mixture inside small diameter microfin copper tubes.

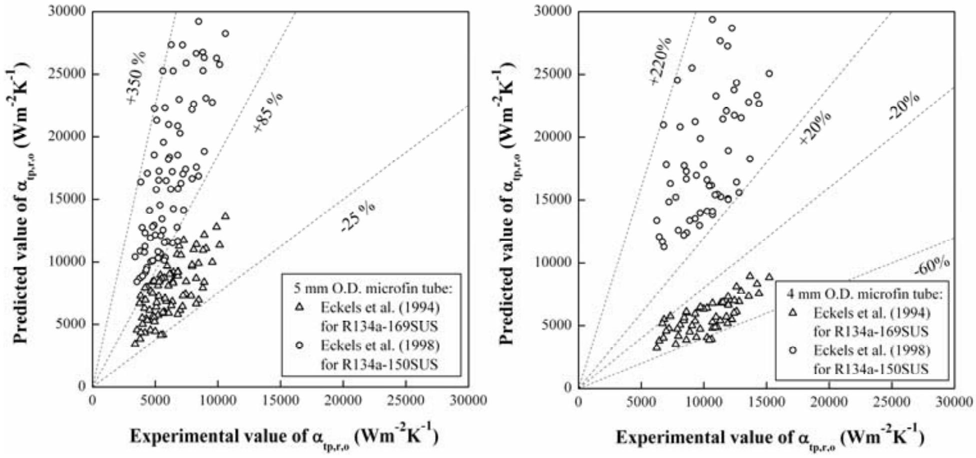
There are two categories of prediction methods: oil enhancement factor based method, and physical properties of refrigerant-oil mixture based method. The predictability verifications of existing prediction methods are illustrated as follows:

(1) The first category of prediction methods employs an oil enhancement factor (*EF*) to correct the two-phase heat transfer coefficient of pure refrigerant, e.g. Schlager et al. correlation [16], Sur and Azer correlation [17], and Eckels et al. correlations [18, 19], etc. Among these correlations, the oil enhancement factor of Schlager et al. correlation and Sur and Azer correlation is defined as the ratio of heat transfer coefficient of refrigerant-oil mixture in microfin tube to that of pure refrigerant in microfin tube; the oil enhancement factor of Eckels et al. correlations is defined as the ratio of heat transfer coefficient of refrigerant-oil mixture in microfin tube to that of pure refrigerant in smooth tube.

Figure 5 shows the deviations of *EF* correlation predictions from the present experimental data of R410A-oil mixture. The predicted values are obtained by the product of predicted *EF* and heat transfer coefficient calculated by pure refrigerant correlation. For predicting the heat transfer coefficients of refrigerant-oil mixture by Schlager et al. correlation and Sur and Azer correlation, Yu and Koyama correlation [20] is chosen to calculate the heat transfer coefficients of pure refrigerant in microfin tubes; while for Eckels et al. correlations, Haraguchi et al. correlation [21] is chosen to calculate the heat transfer coefficients of the pure refrigerant in smooth tubes. It is shown that the deviations of Schlager et al. correlation [16], Sur and Azer correlation [17], Eckels et al. correlation [18], and Eckels et al. correlation [19] are within -10% ~ +40%, -10% ~ +40%, -25% ~ +85% and +85% ~ +350% for 5 mm tube, within -30% ~ +10%, -30% ~ +10%, -60% ~ -20% and +20% ~ +220% for 4 mm tube, respectively. The prediction precision of these correlations can not meet the requirement of engineering application.



(a) Schlager et al. correlation (1990) and Sur and Azer correlation (1991)

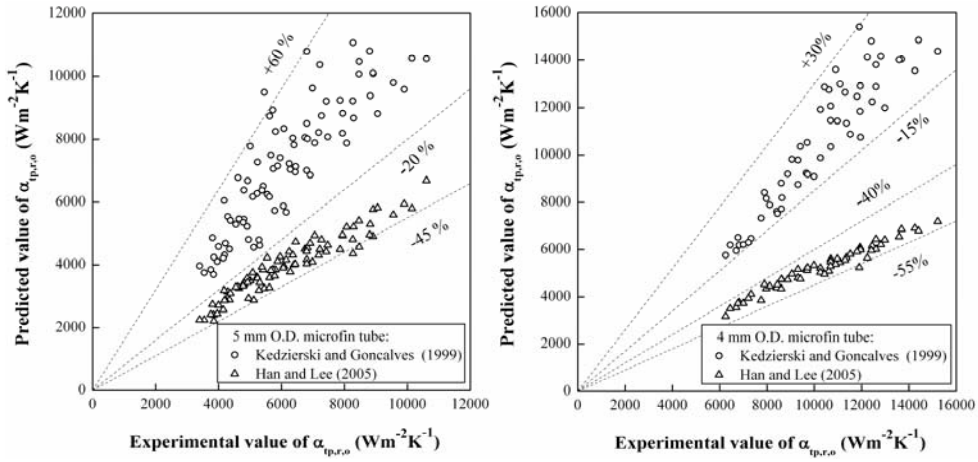


(b) Eckels et al. correlations (1994, 1998)

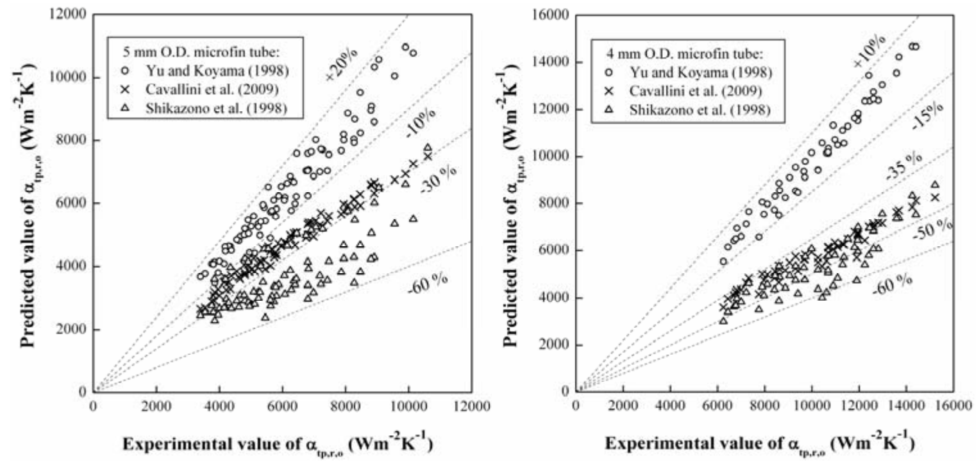
Fig. 5. Comparison of experimental condensation heat transfer coefficients of R410A-oil mixture with the predicted values of *EF* correlations

(2) The second category of prediction methods considers the refrigerant-oil mixture's properties. The common way of the third category is to use a correlation for the pure refrigerant heat transfer coefficient to predict the heat transfer coefficient of the refrigerant-oil mixture by replacing the pure refrigerant properties with the mixture's properties [22]. However, whether the condensation heat transfer characteristics of R410A-oil mixture can be predicted by this method is still uncertain and needs to be verified.

Figure 6 depicts the comparison of the present experimental data with predicted heat transfer coefficients by substituting mixture properties into some classical pure refrigerant correlations. These pure refrigerant correlations include Cavallini et al. correlation [23], Han and Lee correlation [6], Kedzierski and Goncalves correlation [8], Shikazono et al. correlation [24], and Yu and Koyama correlation [20]. The reason for choosing these correlations is that they can provide good prediction precision to the experimental data of pure R410A [6, 25].



(a) Kedzierski and Goncalves correlation (1999) and Han and Lee correlation (2005)



(b) Yu and Koyama correlation (1998), Cavallini et al. correlation (2009)
and Shikazono et al. correlation (1998)

Fig. 6. Comparison of experimental data of R410A-oil mixture with the predicted values obtained by substituting R410A-oil mixture properties into existing pure refrigerant correlations

It can be seen from Fig. 6 that, for 5 mm O.D. tube, the prediction deviations of Cavallini et al. correlation [23], Han and Lee correlation [6], Kedzierski and Goncalves correlation [8], Shikazono et al. correlation [24], and Yu and Koyama correlation [20] are -30% ~ -10%, -45% ~ -20%, -20% ~ +60%, -60% ~ +10% and -10% ~ +20%; while for 4 mm O.D. tube, the prediction deviations of these correlations are -50% ~ -35%, -55% ~ -40%, -15% ~ +30%, -60% ~ -35% and -15% ~ +10%, to the local heat transfer coefficient of R410A-oil mixture, respectively. The conclusion can be deduced that, by substituting the refrigerant-oil mixture's properties, Yu and Koyama correlation provides the best prediction precision to the heat transfer coefficients of R410A-oil mixture flow condensation inside 5 mm and 4 mm O.D. microfin tubes. Therefore, Yu and Koyama correlation is recommended as the heat transfer correlation for R410A-oil mixture in small diameter microfin tubes by using mixture's properties, as tabulated in Appendix A.

6 Conclusions

Small diameter copper tubes are cost effectively in heat exchangers for residential air conditioners compared with aluminum alternatives, and it can reduce 20~30% refrigerant charge and also 20~30%

heat exchanger cost, respectively. Then small diameter copper tube technology is a competitive and economic technology for HVAC industry.

For the purpose of promoting the application of small diameter copper tube, heat transfer characteristics of R410A and ester oil RB68EP mixture flow condensation inside small diameter microfin copper tubes are investigated experimentally. Following conclusions are obtained:

(1) For pure R410A and R410A-oil mixture at 1% nominal oil concentration, the condensation heat transfer coefficient decreases with the decrease of vapor quality; while for R410A-oil mixture at 3% and 5% nominal oil concentrations, a maximum of heat transfer coefficient appears at the vapor quality between 0.7 and 0.75.

(2) The presence of oil deteriorates the flow condensation heat transfer of R410A. At nominal oil concentration of 1%, the deterioration effect is negligible; while at nominal oil concentrations of 3% and 5%, the presence of oil degrades the heat transfer seriously by maximum of 25.1% and 23.8% for 5 mm and 4 mm tubes during high vapor qualities, respectively.

(3) The existing heat transfer coefficient prediction methods of refrigerant-oil mixture flow condensation in microfin tubes are verified with experimental data of the present study, and Yu and Koyama correlation (1998) shows the best predictability. By replacing the pure refrigerant properties with the mixture's properties, Yu and Koyama correlation is recommended as the heat transfer correlation for R410A-oil mixture in small diameter microfin copper tubes.

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APPENDIX A: Yu and Koyama correlation [20]

Author	Fluid	Diameter of test tube	Model
Yu and Koyama correlation (1998)	Pure R123, R134a and R22-oil mixture	6.49 - 8.48 mm I.D.	$\alpha = \varepsilon \mathrm{Nu} \lambda_\mathrm{L} / d_\mathrm{e}$ $\mathrm{Nu} = (\mathrm{Nu}_\mathrm{f}^2 + \mathrm{Nu}_\mathrm{b}^2)^{0.5}$ $\mathrm{Nu}_\mathrm{f} = 0.152(0.3 + 0.1\mathrm{Pr}_\mathrm{L}^{1.1})\mathrm{Re}_\mathrm{L}^{0.68}\phi_\mathrm{V} / X_\mathrm{tt}$ $\mathrm{Nu}_\mathrm{b} = 0.725\left(\frac{d}{d_\mathrm{e}}\varepsilon\right)^{-0.25} H(\delta)\left(\frac{\mathrm{Ga}_\mathrm{L}\mathrm{Pr}_\mathrm{L}}{\mathrm{Ph}}\right)^{0.25}$ $\mathrm{Re}_\mathrm{L} = G(1-x)d_e / \mu_\mathrm{L} \quad d_\mathrm{e} = \sqrt{4A/\pi} \quad \phi_\mathrm{V} = 1.1 + 1.3(\mathrm{Fr}_\mathrm{e} X_\mathrm{tt})^{0.35}$ $\mathrm{Fr}_\mathrm{e} = G / \sqrt{\rho_\mathrm{V}(\rho_\mathrm{L} - \rho_\mathrm{V}) g d_\mathrm{e}} \quad \mathrm{Ga}_\mathrm{L} = g\rho_\mathrm{L}^2 d_\mathrm{e}^3 / \mu_\mathrm{L}^2$ $H(\delta) = \delta + \left[10(1-\delta)^{0.1} - 8.0\right]\sqrt{\delta}(1-\sqrt{\delta})$ $X_\mathrm{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_\mathrm{V}}{\rho_\mathrm{L}}\right)^{0.5}\left(\frac{\mu_\mathrm{L}}{\mu_\mathrm{V}}\right)^{0.1} \quad \mathrm{Ph} = c_\mathrm{pL}\Delta T_\mathrm{s} / h_\mathrm{fg}$ $\beta = \left\{1 + \frac{\rho_\mathrm{V}}{\rho_\mathrm{L}}\left(\frac{1-x}{x}\right)\left[0.4 + 0.6\left(\frac{\frac{\rho_\mathrm{L}}{\rho_\mathrm{V}} + 0.4\frac{1-x}{x}}{1 + 0.4\frac{1-x}{x}}\right)\right]\right\}^{-1}$ ΔT_s is the condensation temperature difference, h_fg is the specific enthalpy of evaporation and g is the gravitational acceleration.

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