AN EXPLICIT ALGEBRAIC SOLUTION OF ADIABATIC CAPILLARY TUBE FLOWS

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Abstract. *Capillary tube flows have been resolved through both numerical and analytical approaches. The former require a reasonable understanding of the governing equations of heat and fluid flow, thermodynamic relations, numerical methods, and computer programming, which is usually considered a burdensome task for most refrigeration and air-conditioning practitioners. Alternatively, algebraic analytical solutions for capillary tube flows have been proposed in the literature to avoid this inconvenience, albeit still requiring iterative loops to calculate the refrigerant mass flow rate. Therefore, this paper presents a first-principles algebraic model to solve adiabatic capillary tube flows which is explicit for mass flow rate and requires a minimum set of thermodynamic relations. Comparisons with more than a thousand experimental data points for adiabatic capillary tube flows showed that the model predicts 91% and almost 100% of all data within ±10% and ±15% error bands, respectively.*

Keywords*: refrigeration, capillary tube, adiabatic flow, algebraic model, experimental validation*

1. INTRODUCTION

A capillary tube is simply a small bore tube connecting the condenser to the evaporator. Liquid refrigerant flows into one end, and expands down to the evaporator pressure. In doing so it meters refrigerant at the desired mass flow rate. A capillary tube appears to be quite simple, but the refrigerant flow inside this component is rather complex. The flow offers several challenges for a phenomenological description: turbulence, phase-change, compressibility and non-equilibrium effects all occur in capillary tube flows. Due to the importance of capillary tubes to the refrigeration industry, models for sizing these components have been extensively proposed in the literature, spanning from empirical correlations (e.g., Wolf *et al.*, 1995; Bansal and Rupasinghe, 1996; Melo *et al.*, 1999a, Melo *et al.*, 1999b; Choi *et al.*, 2004) to first-principles simulation codes (Melo *et al.*, 1992; Seixlack *et al.*, 1996; Chung, 1998; Garcia-Valladares *et al.*, 2002a, 2002b; Bansal and Wang, 2004; Hermes *et al.*, 2007). However, mathematical models have been preferred as the empirical correlations are restricted to the experimental range.

In general, the refrigerant flows in capillary tubes are modeled based on the following key assumptions: (i) the capillary is a straight, horizontal and constant cross-sectional area tube; (ii) the viscous compressible flow is one-dimensional in the axial direction; (iii) the heat diffusion is neglected due to the high (10^4) Peclet number; (iv) the pressure drop at the capillary tube entrance and exit sections is disregarded; (v) the two-phase flow is considered homogeneous, and (vi) the metastable flow is neglected due to its inherent unpredictability.

The governing equations, derived from the mass, momentum and energy conservation principles, can be expressed by the following set of ordinary differential equations (Hermes *et al.*, 2007):

$$G\mathrm{d}v + \mathrm{d}p + fG^2 v\,\mathrm{d}z/2D = 0 \tag{1}$$

$$\mathrm{d}h + G^2 v\mathrm{d}v = 0 \tag{2}$$

The specific volume derivative, dv, is obtained from the following thermodynamic relation,

$$\mathrm{d}v = (\partial v/\partial h)_p\,\mathrm{d}h + (\partial v/\partial p)_h\,\mathrm{d}p \tag{3}$$

For a given mass flux G, there are 3 equations and 4 unknowns (p, h, v, z) and, therefore, one unknown must be chosen as the integration domain. Taking pressure as the integration domain, Eq. (1) to Eq. (3) may be re-arranged as

$$\frac{\mathrm{d}z}{\mathrm{d}p} = -\frac{2D}{fG^2 v}\frac{1+G^2\left[v(\partial v/\partial h)_p + (\partial v/\partial p)_h\right]}{1+G^2 v(\partial v/\partial h)_p} \tag{4}$$

$$\frac{\mathrm{d}h}{\mathrm{d}p} = -\frac{G^2 v(\partial v/\partial p)_h}{1+G^2 v(\partial v/\partial h)_p} \tag{5}$$

Equations (4) and (5) express the tube length and enthalpy variation with respect to the refrigerant pressure for any flow regime, respectively. The boundary conditions are the thermodynamic states at the entrance of the capillary tube (condensing pressure and subcooling degree) and the pressure at the capillary tube exit (evaporating or sonic pressure). It should be noted that there are 3 boundary conditions and only 2 equations; this is so because one boundary condition (i.e., the exit pressure) has to be used for the mass flux calculation.

The solution algorithm requires the numerical integration of Eq. (4) and Eq. (5) in an inner loop using a guessed mass flux, which is iteratively corrected in an outer loop as the flow may be choked at the tube exit. Additionally, the thermodynamic properties, particularly the specific volume, its derivatives $(\partial v/\partial p)_h$ and $(\partial v/\partial h)_p$, and the friction factor have to be calculated at every point of the solution domain.

It is worth noting that the numerical approach is time consuming and requires some programming abilities, which are usually considered a burdensome task for most refrigeration and air-conditioning practitioners. As an alternative, algebraic analytical solutions for capillary tube flows have been proposed in the literature to avoid this inconvenience, albeit still require iterative loops for the calculation of refrigerant mass flow rate.

2. EXISTING ALGEBRAIC SOLUTIONS

Probably the most usual approach employed to size capillary tubes is that based on dimensionless experimental correlations between flow parameters (e.g., mass flow rate, condensing pressure, and subcooling degree) and capillary tube geometry (e.g., inner diameter and length), which provides results that, in general, match experimental data within ±15% error bands (e.g., Zhang, 2005). However, these empirical models have no generality as they are constrained to the experimental range. In order to improve the model generality, theoretical models based on the analytical solution of the first-principles equations have also been proposed in the literature.

Yilmaz and Unal (1996) proposed an algebraic model for predicting the mass flow rate of pure refrigerants and refrigerant mixtures in adiabatic capillary tube flows. The model regards the flow as isenthalpic, thus Eq. (1) to Eq. (3) can be rewritten as,

$$\frac{\mathrm{d}z}{\mathrm{d}p} = -\frac{2D}{fG^2 v}\left[1 + G^2 (\partial v/\partial p)_h\right] \tag{6}$$

Also, the authors assumed a constant specific volume in the subcooled region, thus the length occupied by the subcooled liquid was calculated integrating Eq. (6) from the capillary inlet to the flash-point,

$$L_{sp} = -\int_i^f \frac{2D}{fG^2 v}\,\mathrm{d}p = \frac{2D}{f_{sp}G^2}\frac{p_i - p_f}{v_f} \tag{7}$$

In addition, the authors demonstrated that the two-phase specific volume along an isenthalpic path can be calculated by $v=a+b/p$, where $a=v_f(1\text{-}k)$, $b=v_f p_f k$, $k=2.62\cdot 10^5 p_f^{-0.75}$, $p_f=p_{vap}(h_{liq}=h_i)$, $v_f=v_{liq}(p_f)$, and $(\partial v/\partial p)_h=-b/p^2$. Hence, the two-phase length was calculated integrating Eq. (6) from the flash-point to the capillary tube exit,

$$L_{tp} = -\int_f^e \frac{2D}{fG^2}\left(\frac{1-G^2 b/p^2}{a+b/p}\right)\mathrm{d}p = \frac{2D}{f_{tp}G^2}\left[\frac{p_f - p_e}{a} + \frac{b}{a^2}\ln\left(\frac{ap_e + b}{ap_f + b}\right) - G^2 \ln\left(\frac{v_e}{v_f}\right)\right] \tag{8}$$

Noting that the total tube length is given by $L=L_{sp}+L_{tp}$, Eq. (7) and Eq. (8) can be rewritten for the mass flow rate, w, as

$$w = \frac{\pi D^2}{4}\sqrt{\frac{\dfrac{f_{tp}}{f_{sp}}\dfrac{p_i - p_f}{v_f} + \dfrac{p_f - p_e}{a} + \dfrac{b}{a^2}\ln\left(\dfrac{ap_e + b}{ap_f + b}\right)}{\dfrac{f_{tp}L}{2D} + \ln\left(\dfrac{v_e}{v_f}\right)}} \tag{9}$$

where w is given in [kg/s]. In the work of Yilmaz and Unal (1996), both single and two-phase friction factors were calculated using the correlation presented by Churchill (1977), considering the following averaged viscosity for the two-phase region,

$$\mu_{tp} = \tfrac{8}{7}\mu_f\left[\frac{1-(p_e/p_f)^{7/8}}{1-p_e/p_f}\right] \tag{10}$$

Figure (1) compares the predictions of the formulation introduced by Yilmaz and Unal (1996) with more than a thousand experimental data points gathered at the POLO Laboratories of the Federal University of Santa Catarina for adiabatic capillary tube flows of refrigerants R-12, R-134a, R-22, R404A, R-407C, R-507A and R-600a under several operational and geometric conditions, summarized in Tab. (1). As can be seen, the model predicts 86.0% of all data within ±10% error bands.

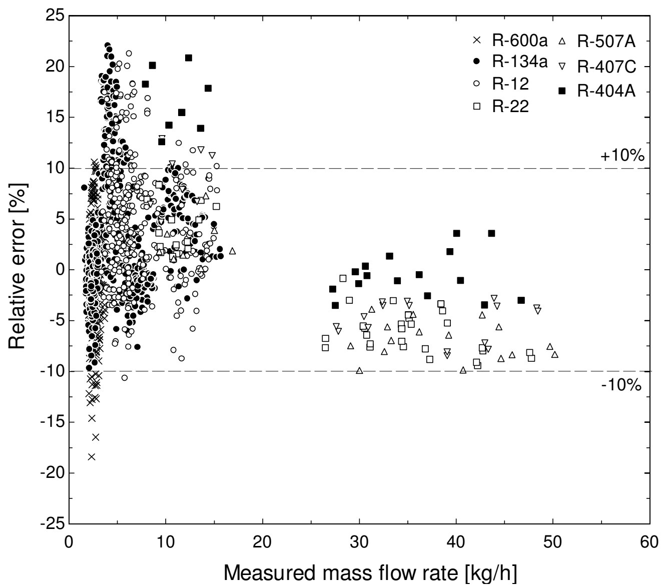

Figure 1. Comparisons of the Yilmaz and Unal (1996) model predictions with experimental data

The original model introduced by Yilmaz and Unal (1996) assumed the exit pressure to be equal to the evaporating pressure, neglecting the possibility of choked flow at the capillary outlet. Zhang and Ding (2004) improved the original formulation by considering the exit pressure to be p_e=max(p_{evap}, p_{sonic}), where the choked flow condition at the capillary exit was obtained setting dz/dp→0 in Eq. (6),

$$p_{sonic} = G\sqrt{v_f\, p_f\, k} \tag{11}$$

Zhang and Ding (2004) added several other contributions to the original formulation. Firstly, they refitted the correlation for $k(p_f)$, proposing $k=1.63\cdot 10^5 p_f^{-0.72}$, which is also applicable for refrigerant mixtures. Also noting that the original formulation was implicit because of the friction factor dependence upon the mass flow rate, the authors proposed a two-step predictor-corrector solution for Eq. (9), using an approximation for the mass flow rate used to calculate the friction factor. Figure (2) compares the predictions of the formulation introduced by Zhang and Ding with all experimental data points of Tab. (1), showing that the model predicts 81.4% of all data within ±10% error bands.

Table 1. Range of experimental data

Fluid	# Points	Key Publication	Range	d	L	p_i	Δt_{sub}	p_e	w
				[mm]	[m]	[bar]	[°C]	[bar]	[kg/h]
R600a	189	Melo *et al.* (1999b)	Max	0.770	2.926	11.31	15.9	1.08	3.90
			Min		2.009	7.13	1.9	0.93	2.01
R134a	572		Max	1.050	3.020	16.63	20.9	2.12	15.70
			Min	0.606	2.009	9.01	1.3	0.92	1.61
R12	288		Max	1.050	3.027	18.34	16.2	2.42	15.33
			Min	0.770	1.993	9.19	1.8	0.68	3.39
R22	48	Melo *et al.* (2004)	Max	1.501	4.000	1951	10.3	504	47.77
			Min	0.993	1.999	1530	4.6	239	8.05
R404A	24		Max	1.495	4.000	23.08	10.3	3.37	46.7
			Min	1.003	1.999	18.28	5.3	2.90	8.61
R407C	24		Max	1.500	4.000	22.25	14.9	3.65	48.3
			Min	1.000	2.000	17.34	10.0	2.42	9.52
R507A	24		Max	1.495	4.000	23.69	9.9	3.69	45.8
			Min	1.003	1.999	18.74	4.7	3.48	9.31

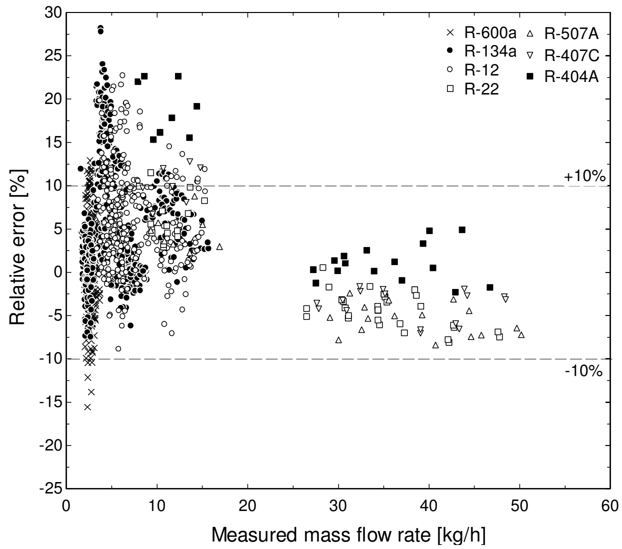

Figure 2. Comparisons of the Zhang and Ding (2004) model predictions with experimental data

It is worth noting that, in both cases, the mass flow rate cannot be calculated straightforwardly by Eq. (9) because of the implicitness of the friction factor. In order to address this issue, Yang and Wang (2007) derived an empirical π-type dimensionless correlation based on the formulation by Zhang and Ding (2004), which is explicit for the mass flow rate. The following correlation was proposed by the authors,

$$\frac{w}{\pi D^2\sqrt{p_i/v_i}} = 1064.5\left(\frac{p_i}{p_f}\right)^{0.7338}\left(\frac{v_{v,i}}{v_{l,i}}\right)^{-0.2220}\left(\frac{D}{L}\right)^{0.4671}\left(\frac{D\sqrt{p_i/v_i}}{\mu_i}\right)^{0.1226}(1-x_i)^{1.5956}\left(1+\frac{\Delta T_{sub}}{T_c}\right)^{0.7061} \tag{12}$$

where the mass flow rate is given in [kg/h]. Figure (3) compares the predictions of Eq. (12) with all experimental data points, showing that the model predicts 68.4% of all data within ±10% error bands. The numerical predictions for the mass flow rate, however, did not show a comparable accuracy to those obtained with the implicit models. To propose an accurate explicit formulation is therefore the main focus of this study.

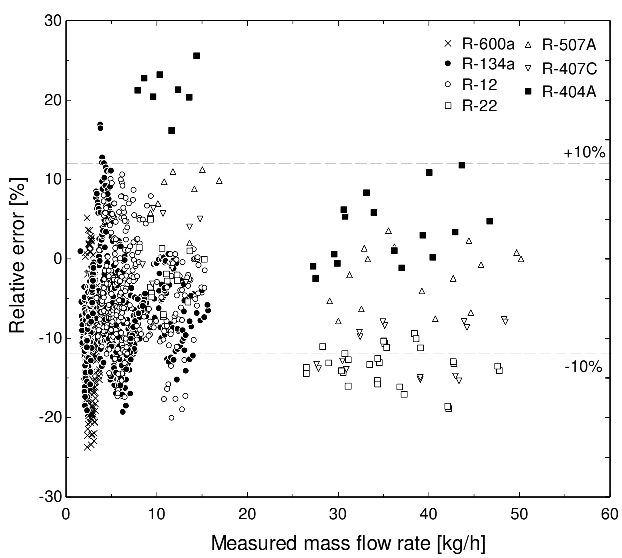

Figure 3. Comparisons of the Yang and Wang (2007) correlation predictions with experimental data

3. PROPOSED EXPLICIT SOLUTION

The formulation introduced in this study was devised considering that: (i) the effect of the mechanical energy variation upon the exit enthalpy is practically negligible (~0.1 kJ/kg), thus the flow can be regarded as isenthalpic (Yilmaz and Unal, 1996); and (ii) the pressure drop due to flow acceleration ($G\mathrm{d}v$) is negligible in comparison to the friction term ($fG^2v\mathrm{d}z/2D$). Looking at Eq. (9), it can be seen that the term $fL/2D$ is at least one order of magnitude higher than the term $\ln(v_e/v_f)$. Figure (4) plots $\ln(v_e/v_f)$ / $fL/2D$ as a function of the Reynolds number ($\mathrm{Re}=GD/\mu_f$) for all experimental data points showing values of the order of -0.1. In this case, f was calculated using Churchill's (1977) correlation.

It is worth noting that dropping the term $G\mathrm{d}v$ out of the momentum equation does not mean that the effect of flow acceleration was neglected since the specific volume is still present in the friction term. Thus, Eq. (6) can be simplified to,

$$\mathrm{d}z = -\frac{2D}{fG^2 v}\mathrm{d}p \tag{13}$$

which may be re-written for the mass flow rate as,

$$w = \frac{\pi}{2\sqrt{2}}\sqrt{\frac{D^5}{fL}\left(-\int_{p_i}^{p_e}\frac{\mathrm{d}p}{v}\right)} \tag{14}$$

Adopting the approximation introduced by Yilmaz and Unal (1996) for the two-phase specific volume, $v=a+b/p$, Eq. (14) may be re-written as,

$$w = \Phi\sqrt{\frac{D^5}{L}\left[\frac{p_i - p_f}{v_f} + \frac{p_f - p_e}{a} + \frac{b}{a^2}\ln\left(\frac{ap_e + b}{ap_f + b}\right)\right]} \tag{15}$$

where w is given in [kg/s], and $\Phi=\pi(8f)^{-1/2}$. Figure (5) plots the experimental values of Φ as a function of Re ($=4w/\pi D\mu_f$) for all experimental data points of Tab. (1). In this case, the two-phase specific volume was calculated using the correlation fitted by Zhang and Ding (2004) ($k=1.63\cdot 10^5 p_f^{-0.72}$).

As can be seen in Fig. (5), most of the Φ-values (~80%) are concentrated within the interval [5.5, 6.5]. Figure (6) shows a histogram with the frequency distribution of Φ, which follows a Gaussian pattern with an average value of 6.08 and a standard deviation of 0.49. It should be noted that assuming $\Phi\approx 6.0$ and approximating the exit pressure by its evaporating counterpart, Eq. (15) becomes explicit for mass flow rate, tube length or capillary diameter, requiring three thermodynamic relations only, $h_{liq}(T_{sat})$, $v_{liq}(p)$, and $p(h_{liq})$.

Figure (7) compares the predictions of Eq. (15) with all experimental data points, showing a reasonable level of agreement, with 85.2% and 92.6% of all data being within ±10% and ±15% error bands, respectively. A global RMS error of 7.6% was also achieved, which is practically the same level of accuracy obtained using the implicit algebraic models of Yilmaz and Unal (1996) and Zhang and Ding (2004).

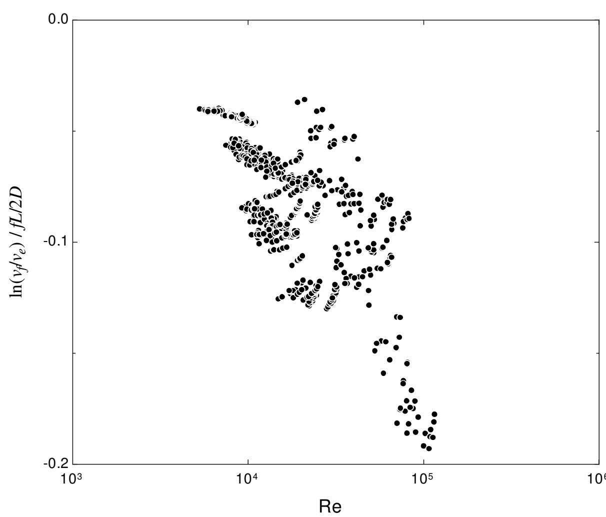

Figure 4. Variation of $\ln(v_f/v_e)$ / $fL/2D$ with Re

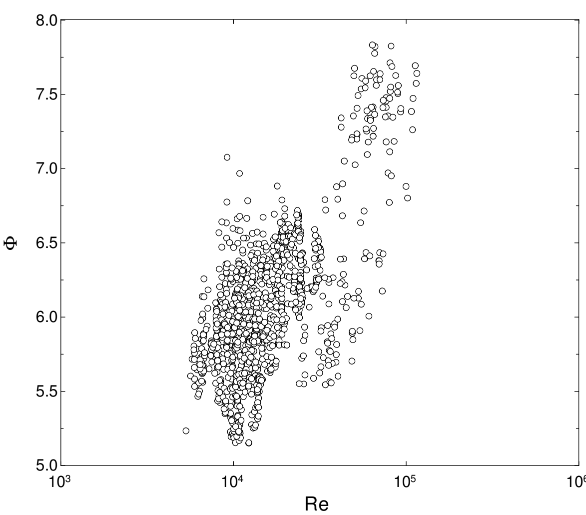

Figure 5. Variation of Φ with Re

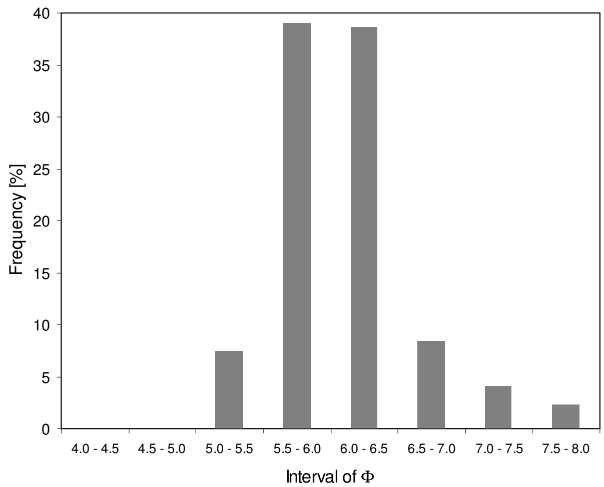

Figure 6. Frequency distribution of Φ

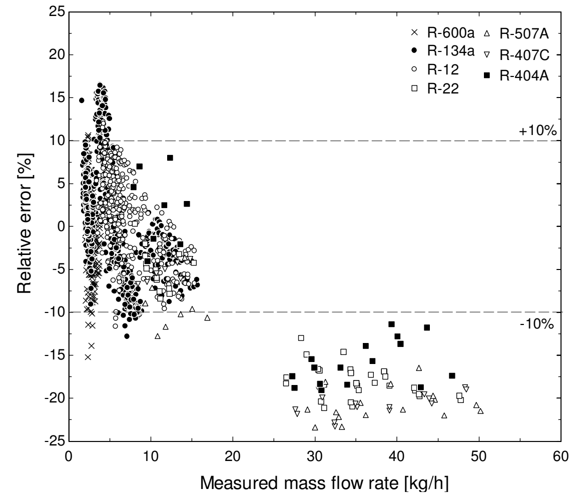

Figure 7. Comparisons of the predictions of Eq. (15) with experimental data

It can also be noted that, if an empirical friction factor $f=c\mathrm{Re}^{-d}$ is used instead of a constant Φ-value, the model predictions improve substantially. Substituting the friction factor equation, $f=c(4m/\pi D\mu_f)^{-d}$, into Eq. (15), the following explicit equation for the mass flow rate through the capillary tube is obtained,

$$w=\left\{\frac{\pi^{2-d}2^{2d-3}}{c}\frac{D^{5-d}}{\mu_f^d L}\left[\frac{p_i-p_f}{v_f}+\frac{p_f-p_e}{a}+\frac{b}{a^2}\ln\left(\frac{ap_e+b}{ap_f+b}\right)\right]\right\}^{1/(2-d)} \tag{16}$$

As shown in Figure (8), the friction factor was fitted using all experimental data points, providing c=0.18 and d=0.17. Hence, $c\pi^{2-d}2^{2d-3}\approx0.23$, $1/(2-d)\approx0.55$, and $5-d\approx4.83$, yielding

$$w=2.93\frac{D^{2.65}}{\mu_f^{0.093}L^{0.55}}\left[\frac{p_i-p_f}{v_f}+\frac{p_f-p_e}{a}+\frac{b}{a^2}\ln\left(\frac{ap_e+b}{ap_f+b}\right)\right]^{0.55} \tag{17}$$

Figure (9) compares the predictions of Eq. (17) with all experimental data points, showing a good level of agreement, with 91.1% and 99.6% of all data being within ±10% and ±15% error bands, respectively. A global RMS error of 6.0% was also observed, which is practically the same level of accuracy obtained using a more sophisticated model (Hermes *et al.*, 2007). It is worth noting that Eq. (17) requires an extra thermodynamic relation for calculating $\mu_f=\mu_{liq}(p_f)$, which is a low price to pay for its better accuracy.

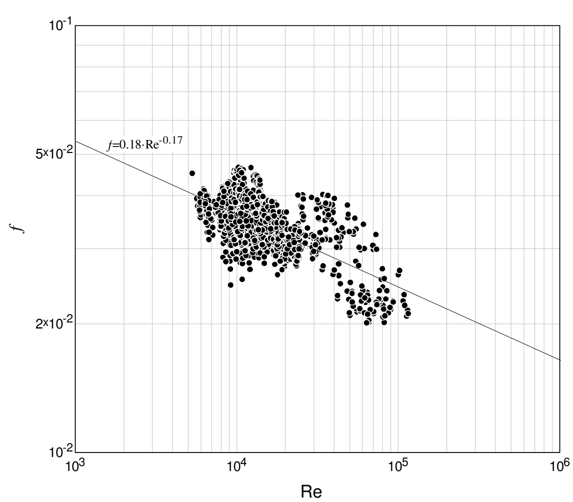

Figure 8. Variation of f with Re for all experimental data

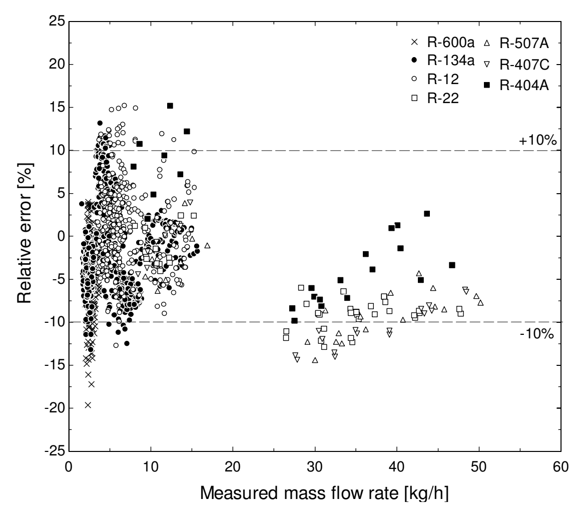

Figure 9. Comparisons of the predictions of Eq. (17) with experimental data

Table (2) compares the accuracy of all the formulations discussed in this paper. It can be seen that Eq. (15), with Φ=6.0, shows a similar accuracy to the implicit formulations, both with RMS errors within 7-8%. In addition, Eq. (17), with c=0.18 and d=-0.17, presents the same accuracy as the distributed model proposed by Hermes *et al.* (2007), both with RMS errors of 6.0%.

Table 2. Error frequency distribution obtained using all methodologies

Formulation	*RMS Error*	*Error frequency* < 10%	*Error frequency* < 15%
	[%]	[%]	[%]
Yilmaz and Unal (1996)	7.1	86.0	93.8
Zhang and Ding (2004)	7.9	81.4	91.9
Yang and Wang (2007)	9.4	68.4	89.2
Hermes *et al.* (2007)	6.0	91.5	97.5
Proposed, Φ	7.6	85.2	92.6
Proposed, *f*	6.0	91.1	99.6

During the experimental work, the diameters of all capillary tubes tested were measured with an uncertainty of ±0.03 mm, which may represent variations up to ±5% in relation to the mean diameters. Since $w \sim D^{2.5}$ in Eq. (15), then $\Delta w/w \sim 2.5\cdot\Delta D/D$, i.e., a 5% variation in the capillary tube diameter may lead to errors in the mass flow rate predictions up to 12.5%, a value of the same order of magnitude as the maximum errors obtained during the model validation exercise. Therefore, model predictions within ±10% error bands can be considered to offer a good level of accuracy.

4. SUMMARY AND CONCLUSIONS

The conclusions of this study can be summarized as follows:

- An explicit algebraic equation was proposed to calculate the refrigerant mass flow rate or tube length or capillary diameter for adiabatic flows;
- The model predictions were compared to a comprehensive experimental dataset for various refrigerants and refrigerant mixtures, predicting 91% of all data within ±10% error bands;
- Equation (15) showed a similar accuracy to the implicit formulations, whereas Eq. (17) presented the same accuracy as that obtained using a distributed model (Hermes *et al.*, 2007);
- The model requires few thermodynamic properties, i.e., only relations for $h_{liq}(T_{sat})$, $v_{liq}(p_{sat})$, $p(h_{liq})$, and $\mu_{liq}(p_{sat})$ are actually needed.

5. NOMENCLATURE

D, inner diameter, m
f, Darcy's friction factor, dimensionless
G, mass flux, kg/m^{2}s
h, specific enthalpy, J/kg
L, capillary tube length, m
p, pressure, Pa
Re, Reynolds number ($=4m/\pi D\mu_f$), dimensionless
T, temperature, K
v, specific volume, m^3/kg
w, mass flow rate, kg/s
z, axial coordinate, m

Greek
Δt_{sub}, subcooling degree at capillary tube inlet, K
μ, viscosity, Pa s
Φ, capillary tube constant (≈6.0), dimensionless

Subscripts
e, capillary tube exit
evap, evaporator
f, flash point
i, capillary tube inlet
liq, saturated liquid
sat, saturation
sonic, choked flow
sp, single-phase
tp, two-phase
vap, saturated vapor

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7. ACKNOWLEDGEMENTS

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