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CAPPILARY TUBE SIMULATION WITH REFRIGERANT MIXTURES FLOW USING SEPARATED FLOW MODEL

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Abstract. *The paper presents a simulation program for refrigerant mixtures flow through capillary tubes using the separated flow model for the two-phase region. The main characteristics of the model are fully described. To perform the model experimental validation an experimental apparatus was built using a blow-down process. Nearly 200 experimental points were obtained for R-407C (a zeotropic mixture) and R-410A (a near azeotropic mixture). Complete data set for each point consists of the measured pressure and temperature profiles, mass flow rate and mixture composition, as well as subcooling/quality control system inlet and outlet temperatures and heater electric power consumption for tests with two-phase flow capillary tube inlet conditions. Comparison of simulation and experimental data show a good agreement. Main deviations are connected with the occurrence of vaporization delay, experimentally verified by the authors. The results are similar to those obtained by another simulation program using homogeneous flow model also developed by the authors.*

Keywords. *Capillary tubes, refrigerant mixtures flow, separated flow model, simulation.*

1. Introduction

Halogenated refrigerants are being replaced by ecologically acceptable alternatives as a consequence of the impositions of Montreal Protocol. One of such substances to be replaced is the HCFC 22, largely used in equipment for commercial refrigeration, commercial and household air conditioners and heat pumps. Unfortunately, to this date, there is no alternative pure substance that could be used without great modifications in existing equipment. The use of zeotropic or near-azeotropic refrigerant mixtures is the most suitable alternative so far. Among the possible alternatives, manufacturers are using mainly R-407C, a zeotropic mixture of HFC 32, HFC 125 and HFC 134a (23% / 25% / 52% on mass basis), R-410A, a near-azeotropic mixture of HFC 32 and HFC 125 (50% / 50% on mass basis) and HFC 134a.

The use of refrigerant mixtures demands experimental and numerical studies in order to evaluate how they affect the performance of refrigeration cycles and design of cycle component. In this way the sizing of adiabatic capillary tubes using zeotropic mixtures is a subject of particular interest. A number of papers can be found in literature (Bittle & Pate, 1996; Sami & Tribes, 1998; Sami et al., 1998; Peixoto et al., 1998; Fiorelli et al., 1998; Jung et al., 1999; Motta et al., 2000) about refrigerant mixtures flow in capillary tubes, all using homogeneous flow for modeling and simulating such refrigeration cycle component. Among them, it should be pointed out the work of Fiorelli & Silvares (2002a), which presented the experimental validation of a simulation model for refrigerant mixtures flow through capillary tubes using the homogeneous flow approach.

Concerning to separated flow model, Wong & Ooi (1996) and Huerta & Silvares (1998) showed, through comparison with literature experimental results, that it could be used, respectively, for pure substances and refrigerant-oil mixtures flow simulation.

In order to verify if the separated flow approach could be used also for refrigerant mixtures flow, this new paper presents a simulation model using this approach. The simulation results are compared to experimental results of Fiorelli et al. (2002), as well as to previous simulation results of Fiorelli & Silvares (2002) for homogeneous flow approach. Such comparison is carried out for several geometries and operational conditions. Calculated and experimental mass flow rates, as well as temperature and pressure profiles, are compared for each mixture and model.

2. Mathematical Model and Simulation Program Implementation

Main assumptions of the model are steady-state one-dimensional flow, pure refrigerant or pure mixture flow (no oil contamination), negligible axial heat conduction in capilary tube walls, constant external UA’ coefficient (heat gain or loss to / from surrounding), no delay of vaporization. Bubble temperatures at condensation and evaporation pressures are assumed as “condensation” and “evaporation” temperatures, respectively. This is made in order to establish a common basis for comparison. Figure 1 shows the variables and parameters involved in capillary tube simulation.

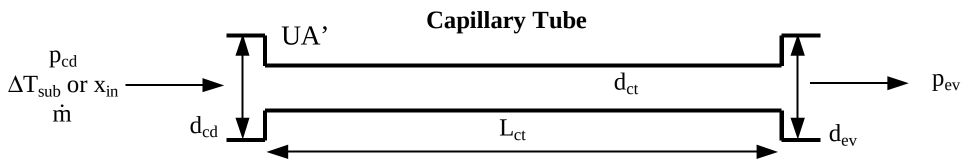


Figure 1. Variables of capillary tubes simulation model

The governing equations are mass, momentum and energy balances, given by Eqs. (1) to (3) for single-phase flow:

$$G = \frac{\dot{m}}{A} = \text{constant} \tag{1}$$

$$\frac{dp}{dz} = -\frac{fvG^2}{2d_{ct}} - G^2\frac{dv}{dz} \tag{2}$$

$$\frac{dh}{dz} = -\frac{h_c \pi d_{ct}}{\dot{m}}\left(T_{ct} - T_w\right) - G^2 v\frac{dv}{dz} \tag{3}$$

The model assumes separated flow model, i.e., liquid and vapor velocities are not equal, and for the two-phase flow region, momentum and energy balances become:

$$\frac{dp}{dz} = -\frac{f_{lo} v_l G^2}{2d_{ct}}\phi_{lo}^2 - G^2\frac{d}{dz}\left[\frac{x^2 v_v}{\alpha} + \frac{(1-x)^2 v_l}{1-\alpha}\right] \tag{4}$$

$$\frac{dh}{dz} = -\frac{h_c \pi d_{ct}\left(T_{ct} - T_w\right)}{\dot{m}} - \frac{G^2}{2}\frac{d}{dz}\left[\frac{x^3 v_v^2}{\alpha^2} + \frac{(1-x)^3 v_l^2}{(1-\alpha)^2}\right] \tag{5}$$

The two-phase multiplier ϕ_{lo}^2 is calculated by Lin's correlation (Lin et al., 1991), Eq. (6), where two-phase Reynolds number Re_{tp} is calculated using a two-phase viscosity presented in Lin's work, Eq. (7):

$$\phi_{lo}^2 = \left\{\frac{\ln\left[\left(7/Re_{lo}\right)^{0.9} + 0.27\varepsilon_{rel}\right]}{\ln\left[\left(7/Re_{tp}\right)^{0.9} + 0.27\varepsilon_{rel}\right]}\right\}\left[1 + x\left(\frac{v_v}{v_l} - 1\right)\right] \tag{6}$$

$$\mu_{tp} = \frac{\mu_l \mu_v}{\mu_v + x^{1.4}\left(\mu_l - \mu_v\right)} \tag{7}$$

The single-phase friction factor f_{lo} evaluated by Serghides correlation (apud Kakaç et al., 1987):

$$\frac{1}{\sqrt{f_{lo}}} = A_5 - \frac{\left(A_5 - B_2\right)^2}{A_5 + 2B_2 + C_1} \tag{8}$$

where

$$A_5 = -0.8686\ln\left(\frac{\varepsilon_{rel}}{7.4} + \frac{12}{Re_{lo}}\right) \tag{9}$$

$$B_2 = -0.8686\ln\left(\frac{\varepsilon_{rel}}{7.4} + \frac{2.51A_5}{Re_{lo}}\right) \tag{10}$$

$$C_1 = -0.8686\ln\left(\frac{\varepsilon_{rel}}{7.4} + \frac{2.51B_2}{Re_{lo}}\right) \tag{11}$$

The critical mass flux is evaluated by (cf. Whalley, 1987):

$$G_{critical}^2 = -\frac{1}{\left[d(MF)/dp\right]_{s=constant}} \tag{12}$$

where MF is the momentum flux, given by Eq. (13):

$$MF = \frac{x^2 v_v}{\alpha} + \frac{(1-x)^2 v_l}{1-\alpha} \tag{13}$$

The void fraction α is given by:

$$\alpha = \frac{x v_v}{1 + S(1-x) v_l} \tag{14}$$

where the slip ratio S is evaluated by Premoli's correlation (Premoli et al., 1971):

$$S = 1 + 1.578\, Re_{lo}^{-0.19} \left(\frac{v_v}{v_l}\right)^{0.22} \sqrt{\frac{A_1}{1 + A_1 F_l} - A_1 F_l} \tag{15}$$

where:

$$A_1 = \frac{\beta}{1-\beta} \tag{16}$$

$$\beta = \frac{x v_v}{x v_v + (1-x) v_l} \tag{17}$$

$$F_l = 0.0273 We_{lo}\, Re_{lo}^{-0.51} \left(\frac{v_v}{v_l}\right)^{-0.08} \tag{18}$$

In order to calculate We_{lo}, surface tension for pure substances and mixtures is evaluated according to relations proposed by Heide (1997). Although Premoli's correlation was originally developed for air-water vertical flow, Friedel (1982), comparing several correlations for slip ratio and void fraction, showed that this correlation also gives the best results for refrigerant horizontal flows.

The capillary tube wall temperature T_w is given by:

$$T_w = \frac{h_c \pi d_{ct} T_{ct} + UA' T_{amb}}{h_c \pi d_{ct} + UA'} \tag{19}$$

where UA' is the overall conductance per unit length of capillary tube. Such conductance includes conduction through capillary tube wall and insulation, as well as external natural convection and radiation. Internal heat transfer coefficient h_c is given, for liquid region, by Dittus-Böelter equation. For the two-phase region a modified Dittus-Böelter equation (cf. Pate, 1982), Eq. (20), with average velocity and liquid properties is used. Exponent n is 0.4 for heating or 0.3 for cooling, and the void fraction α is evaluated by Eq. (14).

$$Nu_l = \frac{h_c d_{ct}}{k_l} = 0.023 Re_l^{0.8} Pr_l^n \left(\frac{1-x}{1-\alpha}\right)^{0.8} \tag{20}$$

Pressure drops at inlet contraction for two-phase flow conditions, as well as at outlet expansion for non-critical flow conditions are calculated according to correlations suggested by Collier & Thome (1996), Eqs. (21) and (22), where $\sigma_{cd} = A_{cd}/A_{ct}$, $\sigma_{ev} = A_{ct}/A_{ev}$ and $C_c = f(\sigma)$. Subcooled liquid inlet pressure drop at inlet is calculated by Eq. (23) (cf. Idelcik, 1960).

$$p_{cd} - p_{in} = \frac{G^2}{2}\left[\left(\frac{1}{C_c} - 1\right)^2 + \left(1 - \frac{1}{\sigma_{cd}^2}\right)\right]\left[\frac{x^2 v_v}{\alpha} + \frac{(1-x)^2 v_l}{1-\alpha}\right] \tag{21}$$

$$p_{out} - p_{ev} = G^2 \sigma_{ev} (1 - \sigma_{ev}) \left[\frac{x^2 v_v}{\alpha} + \frac{(1-x)^2 v_l}{1-\alpha} \right] \tag{22}$$

$$p_{cd} - p_{in} = 1.5 \frac{G^2 v_l}{2} \tag{23}$$

From conservation equations, p(z) and h(z) are obtained along the capillary tube. From these two profiles and overall mixture composition the temperature distribution $T_{ct}(z)$ is achieved. Then, assuming liquid-vapor thermal and hydrodynamic equilibrium, liquid and vapor composition along the capillary tube is calculated. From composition, T(z) and p(z) it is obtained saturated liquid and vapor properties. Finally, x(z) and mixture properties are calculated. Eqs. (24) to (29) show these calculations.

$$T_{ct,i} = f(p_i, h_i, y_1, ..., y_n) \tag{24}$$

$$y_{1,l,i}, ..., y_{n,l,i} = f(p_i, T_i, y_1, ..., y_n) \tag{25}$$

$$y_{1,v,i}, ..., y_{n,v,i} = f(p_i, T_i, y_1, ..., y_n) \tag{26}$$

$$h_{l,i} = f(p_i, x = 0, y_{1,l,i}, ..., y_{n,l,i}) \tag{27}$$

$$h_{v,i} = f(p_i, x = 1, y_{1,v,i}, ..., y_{n,v,i}) \tag{28}$$

$$x_i = \frac{h_i - h_{l,i}}{h_{lv,i}} \tag{29}$$

A computational program was developed for numerical simulation using the EES software (EES, 1997). EES uses an implicit finite difference method for numerical integration of the governing equations and solves the resulting system of nonlinear algebraic equations using a modified Newton-Raphson method. Pressure drop is the step variable. REFPROP routines (NIST, 1996) were used for thermophysical properties evaluation.

Model preliminary experimental validation for pure refrigerants simulation (CFC-12 and HFC-134a) was presented previously (Fiorelli et al., 1998). Calculated mass flow rates agree with experimental and literature data within 10%.

3. Experimental Analysis

Figure 2 presents a flowchart of the experimental apparatus built for the analyses (previously described by Fiorelli et al., 2002). A blow-down process is used, in order to provide an accurate control of the process parameters.

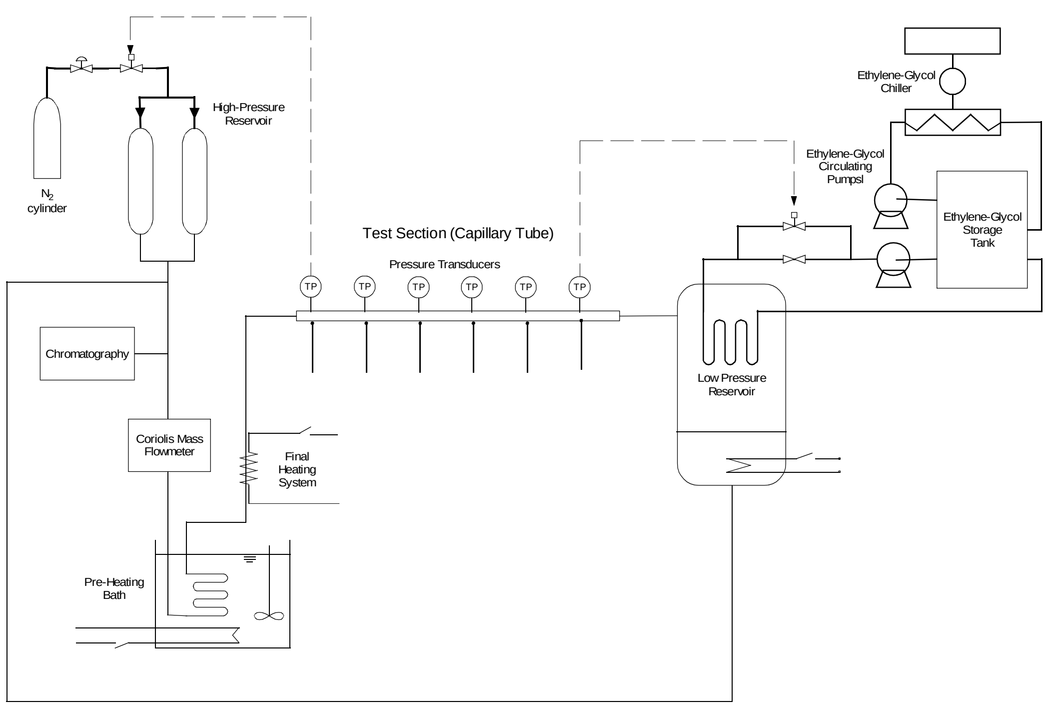


Figure 2. Experimental apparatus flowchart

Ten strain-gage type pressure transducers (± 0.1% range value uncertainty) measure CT pressure profile. Temperature profile is obtained by 18 T-type thermocouples soldered to the tube wall (± 0.3°C uncertainty). It is also used two Pt-100 thermoresistances at capillary tube inlet and subcooling/quality control system inlet (± 0.2°C uncertainty).

Mass flow rate measurement is performed by a Coriolis-type flowmeter (± 0.1% range value uncertainty). Electric power consumption of subcooling/quality final control heater is obtained by a wattmeter (± 1.0% reading uncertainty). This power consumption is used to evaluate, by energy balance, the quality for two-phase capillary tube inlet conditions. Mixture composition (in mass percentage) is monitored by a gas chromatograph with ± 1.0% uncertainty.

Three CT diameters/relative roughness were tested: 1.101 mm ($2.354x10^{-4}$), 1.394 mm ($3.604x10^{-4}$), and 1.641 mm ($2.193x10^{-4}$). CT internal diameter was measured by filling a CT sample with mercury (± 1.0% uncertainty) and relative roughness was evaluated by measuring the pressure losses of an all-liquid R-410A refrigerant flow through CT.

For R-410A it was performed 28 runs, totaling 84 experimental points for a CT diameter d_{ct} = 1.101mm and length L_{ct} = 1.5m. Regarding to R-407C, it was performed 38 runs, totaling 113 experimental points for the following geometries: d_{ct} = 1.101 mm, 1.394 mm and 1.641 mm; L_{ct} = 1.0 m, 1.25 m and 1.5 m. Complete data set for each point consists of the measured pressure and temperature profiles, mass flow rate and mixture composition, as well as subcooling/quality control system inlet and outlet temperatures and heater electric power consumption for tests with two-phase flow capillary tube inlet conditions. Table (1) shows the complete data set for three points of a given run.

Fiorelli et al. (2002) presents the full experimental data set, as well as an experimental analysis of refrigerant mixtures flow through capillary tubes. Such analysis characterized the influence of R-410A and R-407C, as well as the several operating and geometric parameters on the behavior of capillary tubes used in refrigeration systems. It could be seen that the main operational parameters (inlet and outlet conditions) affects in a similar way the performance of capillary tubes for both refrigerants. The study also characterized the influence of geometry (diameter and lenght) on the capillary tubes behavior

Table 2. Example of the Complete Experimental Data Set for a Given Run

Run # 1606 - CT-01 (d_{ct} = 1.101mm; L_{ct} = 1.5m)								
Atmospheric pressure, kPa					93.4			
Mixture composition			% R-32				49.2	
			% R-125				50.8	
Parameter		Position	Point 1		Point 2		Point 3	
			Value	σ^2	Value	σ^2	Value	σ^2
Temperature, °C	PT100-01	n/a	27.5	0.1	30.3	0.1	31.64	0.1
	PT100-02	inlet	31.5	0.1	33.5	0.1	35.51	0.1
	TT-01	0.05	31.4	0.1	33.4	0.1	35.40	0.1
	TT-02	0.15	31.4	0.1	33.4	0.2	35.41	0.1
	TT-03	0.25	31.4	0.1	33.4	0.1	35.34	0.1
	TT-04	0.35	31.4	0.1	33.4	0.1	35.33	0.1
	TT-05	0.45	31.3	0.1	33.3	0.1	33.75	0.1
	TT-06	0.55	31.2	0.1	32.7	0.1	32.03	0.1
	TT-07	0.65	31.0	0.1	30.9	0.1	30.67	0.2
	TT-08	0.75	30.2	0.1	29.6	0.1	29.26	0.1
	TT-09	0.80	29.2	0.1	28.9	0.1	28.55	0.1
	TT-10	0.90	27.6	0.1	27.2	0.1	26.72	0.1
	TT-11	0.95	26.7	0.1	26.3	0.1	25.85	0.1
	TT-12	1.06	24.4	0.1	24.0	0.1	23.37	0.1
	TT-13	1.12	23.2	0.1	22.6	0.1	21.99	0.1
	TT-14	1.18	21.2	0.1	20.6	0.1	19.97	0.1
	TT-15	1.31	16.8	0.1	16.2	0.1	15.54	0.1
	TT-16	1.37	13.7	0.1	13.2	0.1	12.70	0.1
	TT-17	1.43	9.7	0.1	9.3	0.1	8.77	0.1
	TT-18	1.48	4.8	0.1	4.5	0.1	4.19	0.1
Pressure, bar	PT-01	inlet	21.66	0.02	21.67	0.02	21.66	0.02
	PT-02	0.20	20.41	0.03	20.51	0.02	20.58	0.02
	PT-03	0.40	19.56	0.03	19.70	0.02	19.85	0.03
	PT-04	0.50	18.97	0.02	19.17	0.02	19.15	0.02
	PT-05	0.60	18.60	0.02	18.71	0.02	18.60	0.05
	PT-06	0.70	18.02	0.02	17.98	0.02	17.82	0.02
	PT-07	0.85	17.08	0.03	16.94	0.02	16.73	0.02
	PT-08	1.00	15.86	0.02	15.65	0.03	15.40	0.03
	PT-09	1.25	13.13	0.03	12.87	0.03	12.61	0.02
	PT-10	outlet	7.03	0.02	7.03	0.02	7.01	0.02
$\dot{m}_{exp}$, kg/h	FM-01	n/a	25.03	0.17	23.90	0.12	22.93	0.15
$\dot{W}_{HX}$, W	WT-01	n/a	n/a (subcooled inlet)		n/a (subcooled inlet)		n/a (subcooled inlet)	

4. Separated Flow Simulation Program Validation and Comparison to Homogeneous Program

As stated before, the simulation program using the separated flow approach was compared with the above-mentioned experimental data for R-407C and R-410A, as well as with the results of the previously developed simulation program using the homogeneous flow approach. Such comparison is presented in Figs 3 to 6.

4.1. R-410A

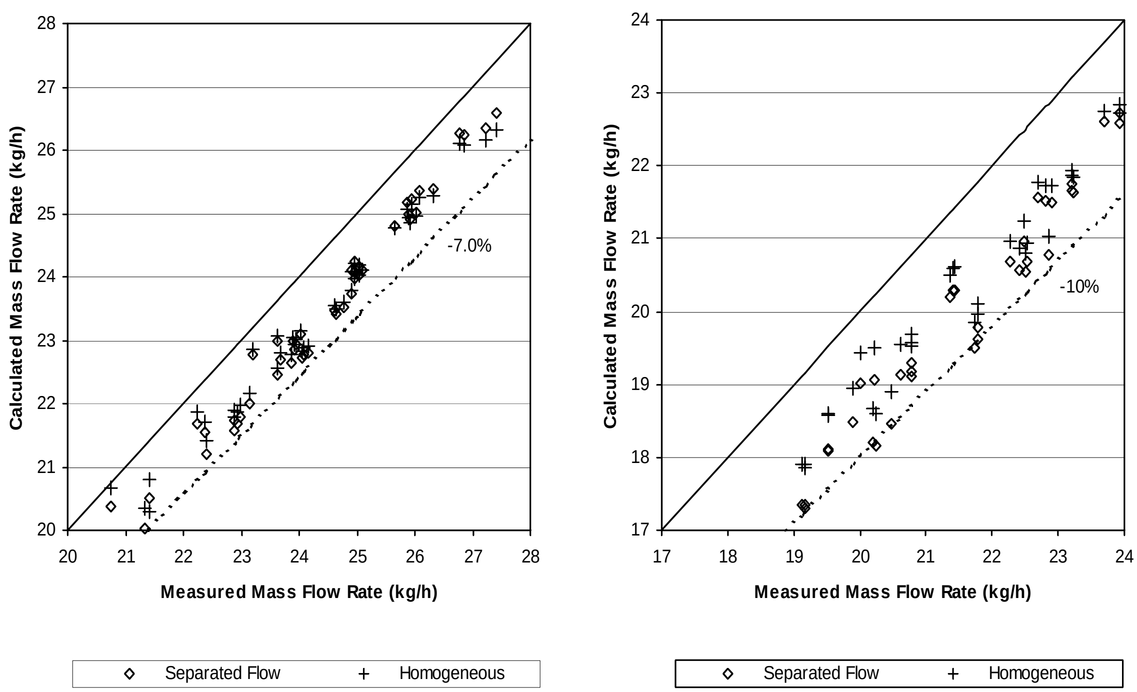
For R-410A, and Figs. 3 to 4, both models underestimate experimental mass flow rate, presenting an average error of -4.0% and a dispersion of ± 2.5% for subcooled inlet. For two-phase inlet, it is obtained an average error of -7.4% for separated flow model and -5.8% for homogeneous model, with a dispersion of ± 3.0% for both models.

It is important to notice that the error increases as subcooling decreases and quality increases. It indicates that, as the two-phase flow region length increases, both models loose accuracy. The larger the length of two-phase region, the bigger is the quality value at capillary tube outlet. So it can be said that for higher quality values, both models become less suitable for representing the physical phenomenon.

One aspect to confirm this fact is that the homogeneous model, which predicts a smaller quality at capillary tube outlet, presents a smaller average error than that of separated flow model for two-phase inlet conditions. On the other hand, for subcooled inlet conditions both models present the same average error (-4.0%).

Figure 4 presents typical measured and calculated pressure and wall temperature profiles for a typical run with subcooled inlet conditions. It can be verified a good agreement between such profiles, with exception of the region where delay of vaporization occurs. This phenomenon is one of the causes that lead to the mass flow rate underestimation. The smaller differences between experimental and simulation results were obtained for those runs where such phenomenon did not occurred. In this sense, the introduction of a prediction model for the delay of vaporization could improve overall capillary tube simulation model. In order to confirm this hypothesis, the phenomenon was taken into account by adding the experimental values obtained for Δp_{sat} (the difference between the inlet saturation pressure and the actual pressure where vaporization begins) to the inlet pressure. This procedure reduced the average error from -4.0% to -1.5% for homogeneous model and from -4.0% to -2.5% for separated flow model. Average values for Δp_{sat} are 36 kPa for R-410A and 50 kPa for R-407C.

Available models in literature to account for this phenomenon were developed for different fluids and geometries ranges and, as expected, did not agree with the data obtained in this study. In this sense, further developments are required.



(a) (b)

Figure 3. Comparison between measured and calculated mass flow rate for R-410A: (a) subcooled and (b) two-phase inlet conditions

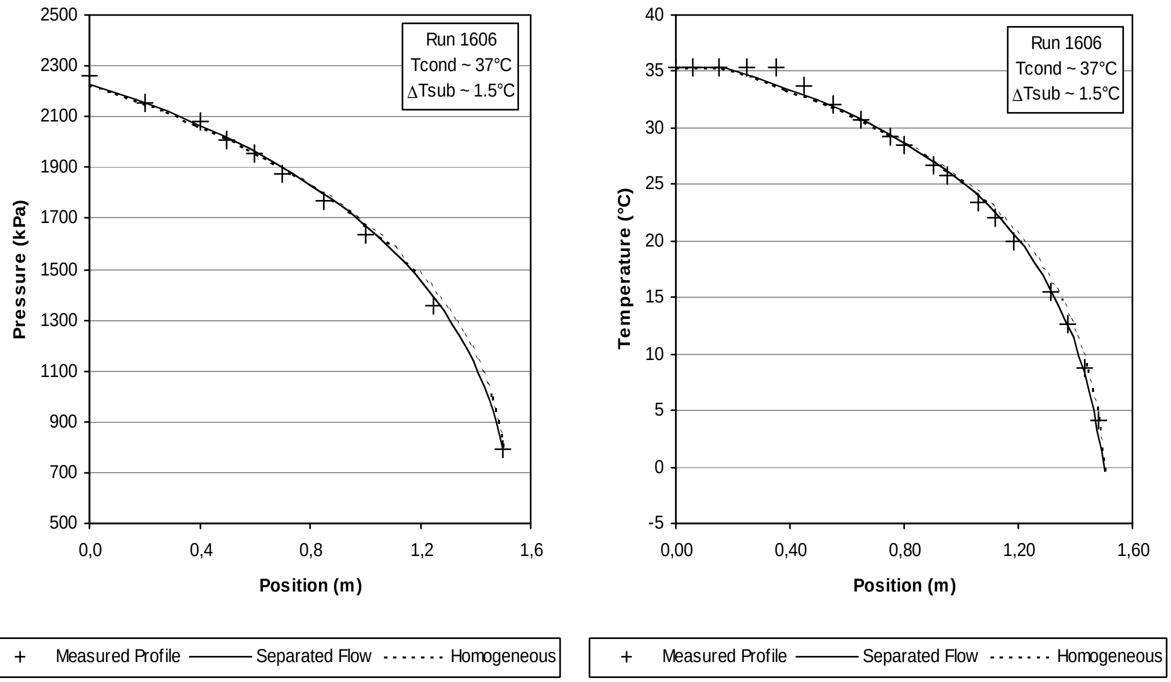


Figure 4. Pressure and wall temperature profiles for R-410A - subcooled inlet conditions

4.2. R-407C

Figures 5 and 6 show the comparison between experimental and calculated data for R-407C. For subcooled inlet, both models present a similar behavior to that for R-410A: they underestimate mass flow rate with an average error of -4.0% and a dispersion of ± 2.5%. However, errors do not present an increase tendency as subcooling decreases.

For two-phase inlet conditions, separated flow model presents an average deviation of -2.8%, whereas homogeneous model presents an average error of +0.1%. In this case, the models behavior is quite different from that for R-410A, once there is no tendency to underestimate mass flow rate.

Agreement between measured and calculated profiles is similar to that for R-410A: there is a good agreement, with exception of the region where delay of vaporization occurs.

The comparison also shows that as capillary tube diameter increases the error between measured and calculated data increases for both models. The same fact is verified for a capillary tube length reduction. Such fact may be connected to the evaluation of critical flow conditions at capillary tube outlet. The separated flow seems to be more adequate than the homogeneous model concerning to this aspect.

5. Conclusion

This paper presents a comparison between two different approaches (homogeneous and separated flow models) for numerical simulation of adiabatic capillary tubes with flow of pure substances and pure refrigerant mixtures. Main assumptions for each model were discussed and governing and constitutive equations were presented.

Experimental validation of simulation programs shows that both models present the same error level when compared to measured values of mass flow rate, temperature and pressure profiles.

The almost identical mass flow rate predictions for the two models seem to be connected to the small cross section area of the capillary tube and the high refrigerant velocity on both liquid and vapor phases, which reduce the effect of the slip ratio between the two phases and approaches the separated flow model to the homogeneous model.

Experimental validation also shows the occurrence of the delay of vaporization, which leads to a mass flow rate underestimation tendency for both models. Taking it into account reduced error level for both models.

The presented results show that both models are suitable for simulation of capillary tubes with refrigerant mixtures flow. The slight differences observed are not significant to determine that one model is better than the other. This conclusion is similar to those of Wong & Ooi (1996) for pure substances flowing through capillary tubes.

6. Acknowledgements

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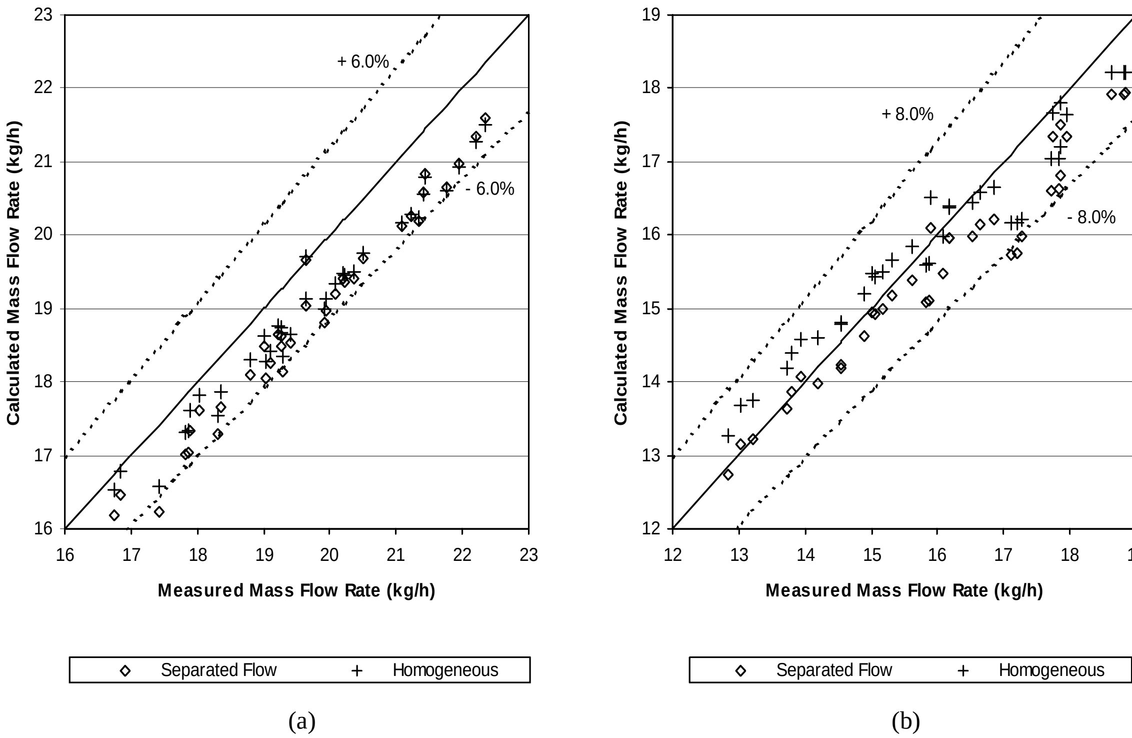

Figure 5. Comparison between measured and calculated mass flow rate for R-407C: (a) subcooled and (b) two-phase inlet conditions

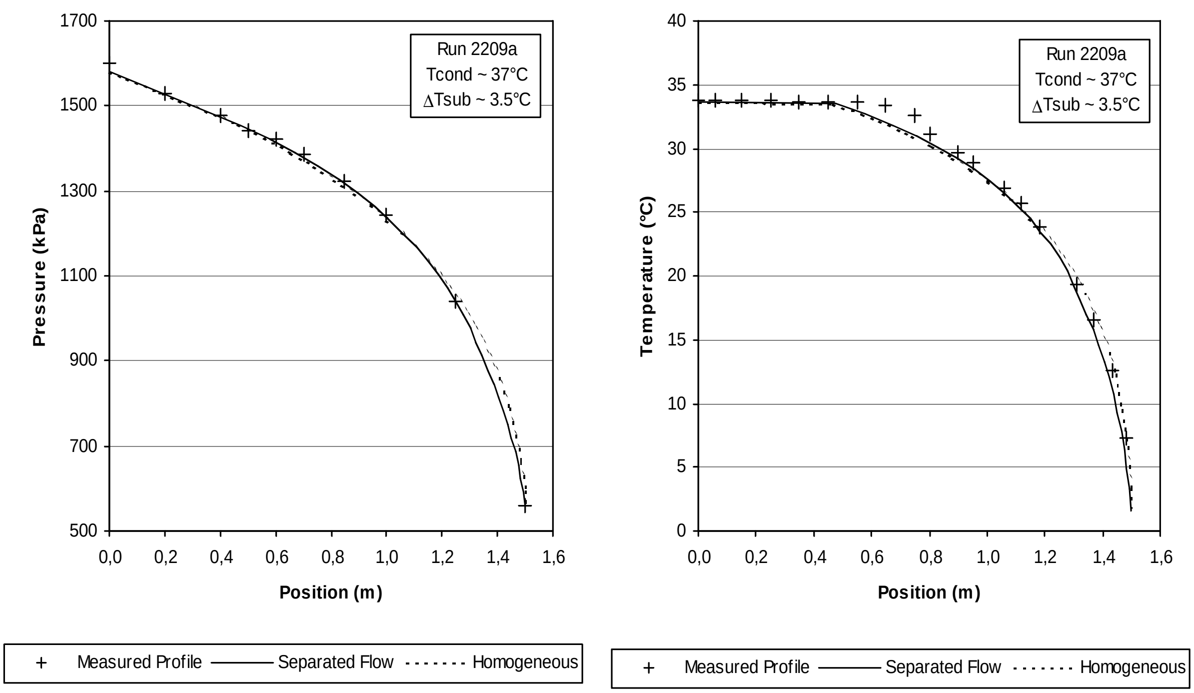

Figure 6. Pressure and wall temperature profiles for R-407C - subcooled inlet conditions

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