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STUDY ON THE BEHAVIOUR OF CAPILLARY TUBES WITH REFRIGERANT MIXTURES FLOW USING A SEPARATED FLOW MODEL

Flávio A. S. Fiorelli
Universidade de São Paulo - Depto. Engenharia Mecânica
Av. Prof. Mello Moraes 2231 – CEP 05508-900 - São Paulo (SP) - Brazil
Roberto A. Peixoto
Instituto Mauá de Tecnologia - Escola de Engenharia Mauá
Marco A. S. Paiva
Instituto de Pesquisas Tecnológicas - SP
Instituto Mauá de Tecnologia - Escola de Engenharia Mauá
Otávio M. Silvares
Universidade de São Paulo - Depto. Engenharia Mecânica
Instituto Mauá de Tecnologia - Escola de Engenharia Mauá

Abstract*. This paper presents a numerical study on the performance of capillary tubes with R-22, R-407C and R-410A using a separated flow model, experimentally validated using data from previous work and literature. It is compared the mass flow rate as function of inlet conditions (condensing temperature and pressure, subcooling degree or inlet quality) for a given geometry. Numerical results show that, taking R-22 as the reference, R-407C mass flow rates are slightly higher (about 5%) and R-410A presents about 30% higher mass flow rates for a fixed condensing temperature. On the other hand, for a fixed condensing pressure, both R-407C (about 3%) and R-410A (from 1% to 5%) mass flow rates are lower than R-22 ones.*

Keywords: Capillary tubes, Refrigerant mixtures, HCFC Replacement

1. INTRODUCTION

The main goal of the Montreal Protocol is the elimination of halogenated compounds. One of such substances is the HCFC 22, largely used as refrigerant in equipment for commercial refrigeration, commercial and household air conditioners and heat pumps. Unfortunately up today there is no pure substance that could be used as alternative without the need of major modifications in existing equipment. The use of zeotropic or near-azeotropic refrigerant mixtures is the most suitable alternative so far.

A recent survey (IIR, 1998) shows that compressor manufacturers as well as manufacturers of small and medium size air conditioning units are using mainly R-407C (a zeotropic mixture with 23% of R-32, 25% of R-125 and 52% of R-134a on mass basis) and R-134a. The use of R-410A (a near-azeotropic mixture with 50% of R-32 and 50% of R-125 on mass basis), which is another possible alternative mixture, has not increased as expected, although Japanese manufacturers are planning to use it in small size units.

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.

So far there are few works in literature (Bittle et al., 1996; Sami & Tribes, 1998; Sami et al., 1998; Peixoto et al., 1998; Fiorelli et al., 1998) about refrigerant mixtures flow in capillary tubes, all using homogeneous flow for modelling and simulating such refrigeration cycle component. Concerning separated flow models, Wong & Ooi (1996) and Huerta & Silvares (1998) showed, through comparison with experimental results, that the model could be used for pure substances and refrigerant-oil mixtures. So far, from authors' knowledge, there is no work about modelling pure refrigerant mixture flow with separated flow model.

This paper presents a comparative numerical study, using a separated flow model, on the performance of HCFC 22 and some alternative refrigerant mixtures flowing through capillary tubes. The performance of an adiabatic capillary tube using HCFC 22 is compared to that with R-407C and R-410A. Mass flow rates are compared as well as quality, void fraction and friction factor profiles for each refrigerant for a given geometry and different operating conditions.

2. MATHEMATICAL MODEL

The mathematical model is based on previous works of the authors for adiabatic and non-adiabatic capillary tubes simulation (Paiva et al., 1995; Peixoto, 1995; Peixoto et al., 1998; Fiorelli et al., 1998) using homogeneous flow hypothesis. The main assumptions of the present model are: steady state one dimensional flow, pure refrigerant or pure refrigerant mixture flow (no oil contamination), negligible axial and radial heat conduction in CT walls, constant external *UA'* coefficient (heat gain or loss from/to ambient), no delay of vaporisation (equilibrium), and separated two-phase flow model. For non-azeotropic and near-azeotropic mixtures it is assumed as "condensation" and "evaporation" temperatures the bubble temperatures at condensation and evaporation pressures. This assumption was made in order to establish a common basis for comparison. The governing equations are mass, momentum and energy balances, given by Eqs. (01) to (03) for two-phase flow region:

$$G = \dot{m}/A = const., \tag{01}$$

$$\frac{dp}{dz} = -\frac{f_{lo} v_l G^2}{2.d_{ct}} \phi_{lo}^2 - G^2 \frac{d}{dz}\left[\frac{x^2 v_v}{\alpha} + \frac{(1-x)^2 v_l}{1-\alpha}\right], \tag{02}$$

$$\dot{m}\frac{dh}{dz} = -h_c \pi d_{ct}\left(T_{ct} - T_w\right) - \dot{m}\frac{G^2}{2}\left[\frac{x^3 v_v^2}{\alpha^2} + \frac{(1-x)^3 v_l^2}{(1-\alpha)^2}\right], \tag{03}$$

where G is the mass flux, $\dot{m}$ the mass flow rate and A the capillary tube (CT) internal cross area; p is the pressure, z the position along the CT, f_{lo} a single-phase Darcy friction factor based on total mass flow rate and liquid properties, v is the specific volume, d_{ct} is the CT diameter, x is the quality and α the void fraction. Finally, h is the enthalpy, T_{ct} is the refrigerant temperature, T_w the CT wall temperature, and h_c is the internal convection heat transfer coefficient. Thermodynamic properties are calculated using REFPROP (NIST, 1996).

Two-phase multiplier ϕ_{lo}^2 in Eq. (02) is calculated by Li's correlation (Li et al., 1991):

$$\phi_{lo}^2 = \left\{ \frac{\ln\left[(7/\mathrm{Re}_{lo})^{0,9} + 0,27\varepsilon_{rel} \right]}{\ln\left[(7/\mathrm{Re}_{tp})^{0,9} + 0,27\varepsilon_{rel} \right]} \right\} \left[1 + x\left(\frac{v_v}{v_l} - 1 \right) \right], \tag{04}$$

where ε_{rel} is the CT relative roughness, Re_{lo} is the Reynolds number based on total mass flow rate and liquid properties and Re_{tp} a two-phase Reynolds number calculated with a two-phase viscosity evaluated by:

$$\mu_{tp} = \frac{(\mu_l \mu_v)}{\left[\mu_v + x^{1,4} (\mu_l - \mu_v) \right]}. \tag{05}$$

Single-phase friction factor f_{lo} is evaluated using Serghides correlation (apud Kakaç et. al., 1987):

$$\left(1/\sqrt{f_{lo}} \right) = A_5 - \frac{(A_5 - B_2)^2}{(A_5 + 2B_2 + C_1)}, \tag{06}$$

where: $A_5 = -0,8686 \ln\left[(\varepsilon_{rel}/7,4) + (12/\mathrm{Re}_{lo}) \right]$,

$B_2 = -0,8686 \ln\left[(\varepsilon_{rel}/7,4) + (2,51 A_5/\mathrm{Re}_{lo}) \right]$,

$C_1 = -0,8686 \ln\left[(\varepsilon_{rel}/7,4) + (2,51 B_2/\mathrm{Re}_{lo}) \right]$.

In order to evaluated critical flow conditions, critical mass flux G_c is evaluated by (cf. Whalley, 1987):

$$G_c^2 = -\frac{1}{\left[\frac{d(MF)}{dp} \right]_{s=const.}}, \tag{07}$$

where the specific momentum flux MF is calculated by:

$$MF = \frac{x^2 v_v}{\alpha} + \frac{(1-x)^2 v_l}{(1-\alpha)}. \tag{08}$$

Void fraction α is calculated by:

$$\alpha = \frac{x v_v}{x v_v + S(1-x) v_l}, \tag{09}$$

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

$$S = 1 + 1,578\, \mathrm{Re}_{lo}^{-0,19} \left(\frac{v_v}{v_l} \right)^{0,22} \sqrt{\frac{A_1}{1 + A_1 F_1} - A_1 F_1}, \tag{10}$$

where: $A_1 = \frac{\beta}{(1-\beta)}$,

$\beta = x v_v / \left[x v_v + (1-x) v_l \right]$, and $F_1 = 0{,}0273\mathrm{We}_{lo}\,\mathrm{Re}_{lo}^{-0{,}51}(v_v/v_l)^{-0{,}08}$.

In Eq. (10), β is the volumetric quality and We_{lo} the Weber number based on total flow and liquid properties. In order to calculate We_{lo}, evaluation of surface tension σ for pure substances and mixtures was made according to Heide (1997).

CT wall temperature T_w in Eq (03) is calculated by:

$$T_w = \frac{h_c \pi d_{ct} T_{ct} + UA' T_{amb}}{h_c \pi d_{ct} + UA'} \,. \tag{11}$$

Internal convection heat transfer coefficient h_c is given, for liquid region, by Dittus-Böelter equation. For two-phase region it is used a modified Dittus-Böelter equation, Eq. (12) (apud Pate, 1982), based on average velocity and liquid properties. Exponent n is 0,4 for heating or 0,3 for cooling.

$$(h_c d_{ct})/k_l = 0{,}023\,\mathrm{Re}_{lo}^{0{,}8}\,\mathrm{Pr}_l^{n}\left[(1-x)/(1-\alpha)\right]^{0{,}8} \tag{12}$$

Pressure drops at inlet contraction for two-phase flow conditions, as well as at outlet expansion for non-critical flow conditions are calculated by Eqs. (13) and (14) respectively (cf. Collier & Thome, 1996), where $\sigma_{cd} = A_{cd}/A_{ct}$ for inlet or $\sigma_{ev} = A_{ct}/A_{ev}$ for outlet, and $C_c = f(\sigma)$. For subcooled liquid inlet conditions pressure drop at inlet is calculated by Eq. (15) (cf. Idelcik, 1960).

$$p_{cd} - p_{in} = \frac{G^2 v_l}{2}\left[\left(\frac{1}{C_c}-1\right)^2 + \left(1-\frac{1}{\sigma_{cd}^2}\right)\right]\left[\frac{(1-x)^2}{1-\alpha} + \left(\frac{v_v}{v_l}\right)\frac{x^2}{\alpha}\right] \tag{13}$$

$$p_{out} - p_{ev} = G^2 v_l \sigma_{ev}(1-\sigma_{ev})\left[\frac{(1-x)^2}{1-\alpha} + \left(\frac{v_v}{v_l}\right)\frac{x^2}{\alpha}\right] \tag{14}$$

$$p_{cd} - p_{in} = 1{,}5\left(G^2 v/2\right) \tag{15}$$

From conservation equations p and h are obtained along the CT. From these two profiles and overall mixture composition a T distribution is achieved. Then, from p and T as well as assuming liquid-vapour thermal and hydrodynamic equilibrium, liquid and vapour molar compositions y_l and y_v along the CT is calculated. From composition, T and p it is obtained saturated liquid and vapour properties. At last, from h it is calculated x profile and mixture properties. Eqs. (16) to (21) show these calculations.

$$T_{ct,i} = f(p_i, h_i, y_1,\ldots, y_n) \tag{16}$$

$$y_{1,l,i},\ldots, y_{n,l,i} = f(p_i, T_i, y_1,\ldots, y_n) \tag{17}$$

$$y_{1,v,i},\ldots, y_{n,v,i} = f(p_i, T_i, y_1,\ldots, y_n) \tag{18}$$

$$h_{l,i} = f(p_i, x=0, y_{1,l},\ldots, y_{n,l}) \tag{19}$$

$$h_{v,i} = f(p_i, x=1, y_{1,v},\ldots, y_{n,v}) \tag{20}$$

$$x_i = (h_i - h_{l,i})/(h_{v,i} - h_{l,i}) \tag{21}$$

3. MODEL IMPLEMENTATION AND VALIDATION

A computational program was developed for numerical simulation using the EES software (EES, 1997). It uses an implicit finite difference method for numerical integration of the governing equations and solves the resulting system of non-linear algebraic equations using a modified Newton-Raphson method. The step variable used is the pressure drop.

Preliminary experimental validation of model was performed with R-134a data from previous work of authors (Paiva et al., 1995) and is shown in Fig. 1. In such figure it is also shown the numerical results obtained with previous homogeneous model from the authors (Fiorelli et al., 1998). It can be seen that both separated flow and homogenous model present results within the range of ±7,5% of experimental data, showing that both a simpler (homogeneous) and a more complicated flow model (separated flow) can be used with the same error level

The model was also compared with experimental data of Sami & Tribes (1998) and Sami et al. (1998) for a capillary tube with d_{ct} = 1,9mm and ε_{rel} = 0,013. Results in Table 1 show that the model presents a fair agreement with experimental data for both binary and ternary mixtures. Fig. 2 compares predicted and measured pressure profiles for R-410A and R-407C, confirming the results of Table 1. Unfortunately the two papers do not mention measurement uncertainties, without which a more complete verification of model accuracy is difficult.

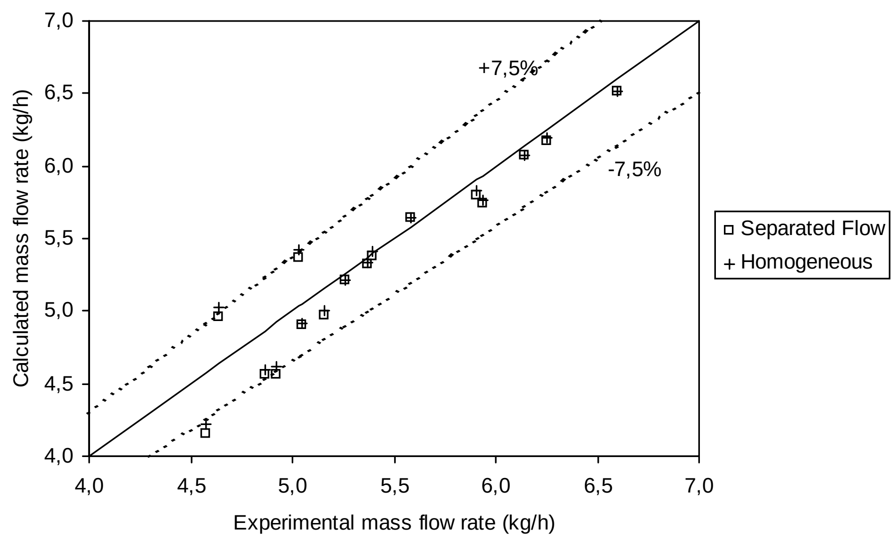

Figure 1: Validation of separated flow model and comparison with homogeneous model

Table 1: Experimental validation of separated flow model

Case	Fluid	T_{in}	$\Delta T_{sub,in}$ / x_{in}	$\dot{m}_{exp}$	$\dot{m}_{calc}$	Deviation
1	R-410A	30,5°C	1,0°C	57,89 kg/h	58,84 kg/h	+1,65%
2	R-410A	34,8°C	0,019	57,96 kg/h	56,56 kg/h	-2,41%
3	R-407C	31,6°C	0,9°C	41,40 kg/h	47,95 kg/h	+15,8%
4	R-407C	31,5°C	0,0°C	46,08 kg/h	46,13 kg/h	+1,09%
5	R-407C	33,0°C	0,02	40,82 kg/h	43,72 kg/h	+7,10%
6	R-407D	26,3°C	0,86°C	38,74 kg/h	39,37 kg/h	+1,63%

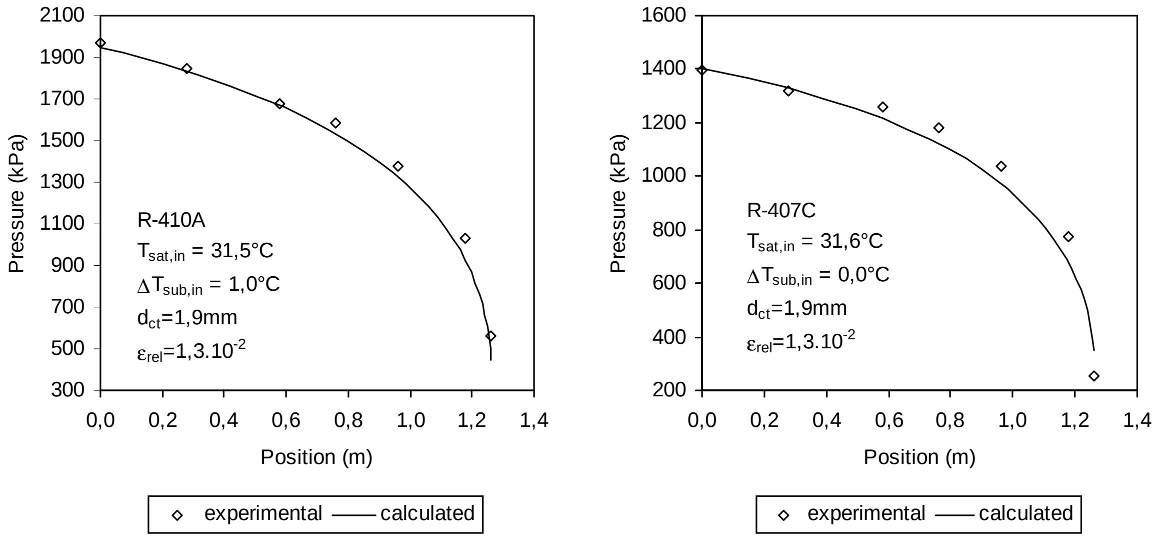

Figure 2. Comparison of predicted and measured pressure profiles for two cases of Table 1

4. NUMERICAL RESULTS

It was developed a numerical study comparing R-22 capillary tube behaviour with R-410A and R-407C refrigerant mixtures. Calculations were performed for the following geometry: d_{ct} = 1,676 mm, L_{ct} = 1,524 m, d_{cd} = d_{ev} = 6 mm. It was assumed constant ambient temperature T_{ext} = 25°C and UA' = 0,11 W/m.°C. This UA' value was experimentally evaluated by Paiva et al. (1995). It was also assumed constant evaporation temperature T_{ev} = -10°C or critical flow.

Figures 3 and 4 show a comparison of mass flow rates for each fluid considering 40°C condensing temperature. Taking R-22 as the reference, R-407C mass flow rates are slightly higher (about 5%) than R-22 and R-410A flow rates are about 30% higher. These results are similar to those obtained previously with the homogeneous model (Fiorelli et al., 1998), and, as stated before, are primarily connected with different saturation pressure curves for each fluid, as shown in Fig. 5.

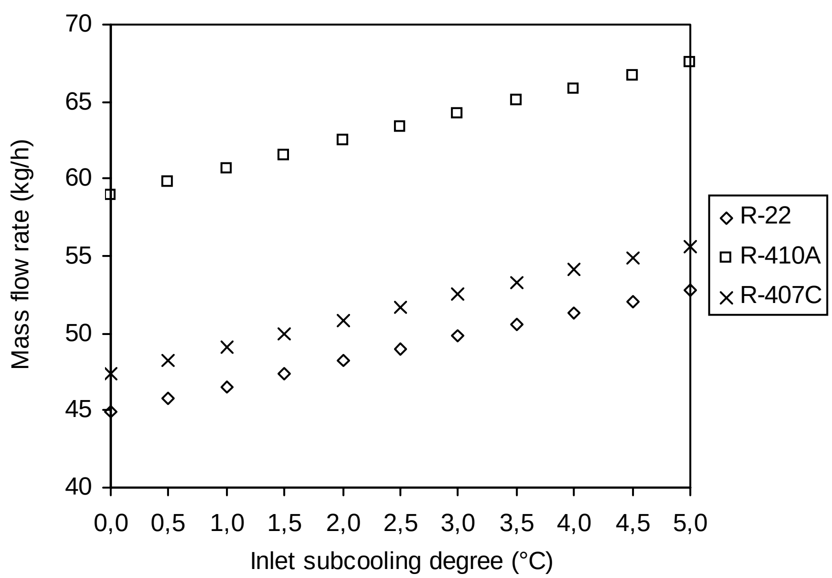

Figure 3: Mass flow rate as function of $\Delta T_{sub,in}$ for 40°C condensing temperature

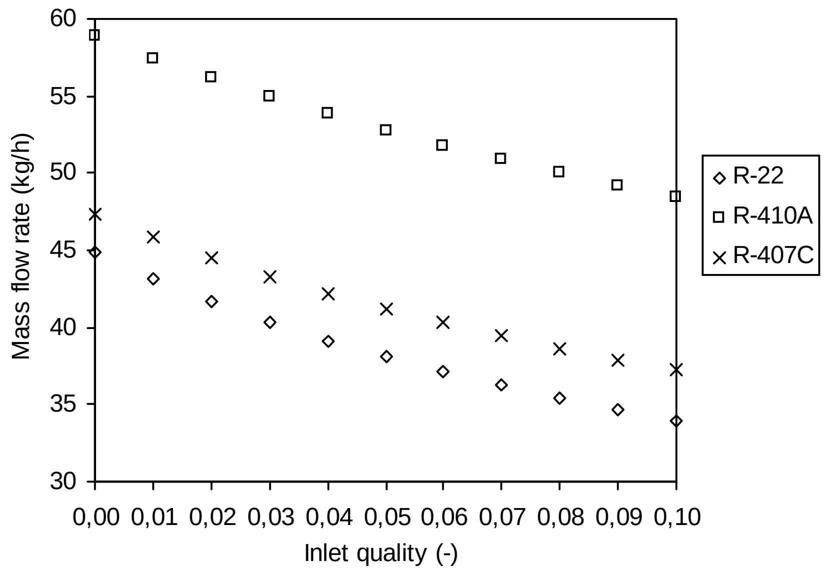

Figure 4: Mass flow rate as function of x_{in} for 40°C condensing temperature

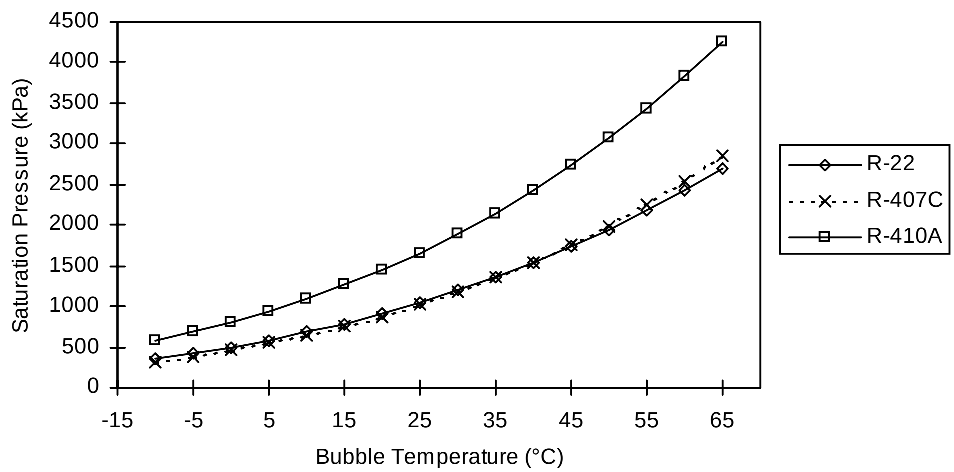

Figure 5: Saturation pressure as function of temperature for R-22, R-407A and R-410A

It was also compared the mass flow rates for the same condensing pressure (1500 kPa), and the results are shown in Figs. 6 and 7. In such case, R-22 mass flow rates are slightly higher than those for R-407C (about 3%) and increasingly higher than those for R-410A (from 1% to 5%) as inlet subcooling degree decreases and inlet quality increases.

Figures 8 to 10 show comparisons of the quality, void fraction and single-phase friction factor profiles for each fluid, considering a 1500 kPa condensing temperature and a fixed mass flow rate (50,0 kg/h). From these figures it can be seen that R-22 CT length is the biggest one of the considered fluids.

5. CONCLUSION

This paper presents a numerical simulation model of adiabatic capillary tubes with flow of pure substances and pure refrigerant mixtures using a separated flow model. Model comparison with a previous homogeneous model from the authors shows that both models can be used to simulate capillary tubes with the same error level for pure substances. The model was also compared with experimental results from literature for refrigerants mixtures. Results in Table 1 and Fig. 2 show that the model presents a fair agreement with experimental data for both binary and ternary mixtures.

Finally it was performed a comparative study of R-22 and their alternatives R-410A and R-407C capillary tube performance. Results show that, taking R-22 as the reference, R-407C mass flow rates are slightly higher (about 5%) and R-410A presents about 30% higher mass flow rates for a fixed condensing temperature. On the other side, for a fixed condensing pressure, both R-407C (about 3%) and R-410A (from 1% to 5%) mass flow rates are lower than R-22 ones. It is also compared some profiles for each fluid considering a given geometry and operational condition.

6. ACKNOWLEDGEMENTS

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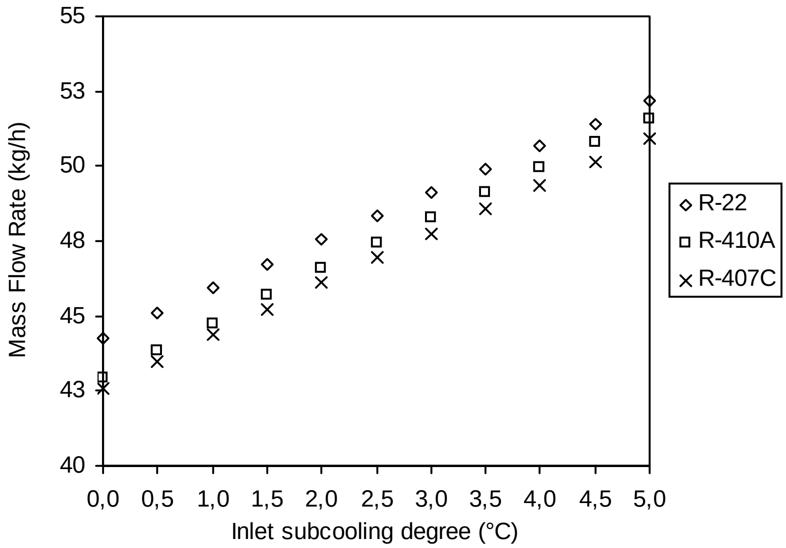

Figure 6: Mass flow rate as function of $\Delta T_{sub,in}$ for 1500 kPa condensing pressure

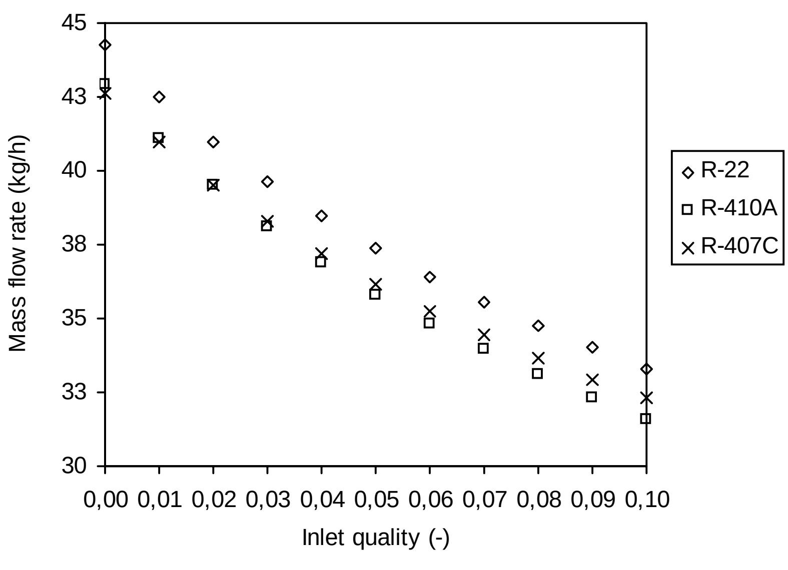

Figure 7: Mass flow rate as function of x_{in} for 1500 kPa condensing pressure

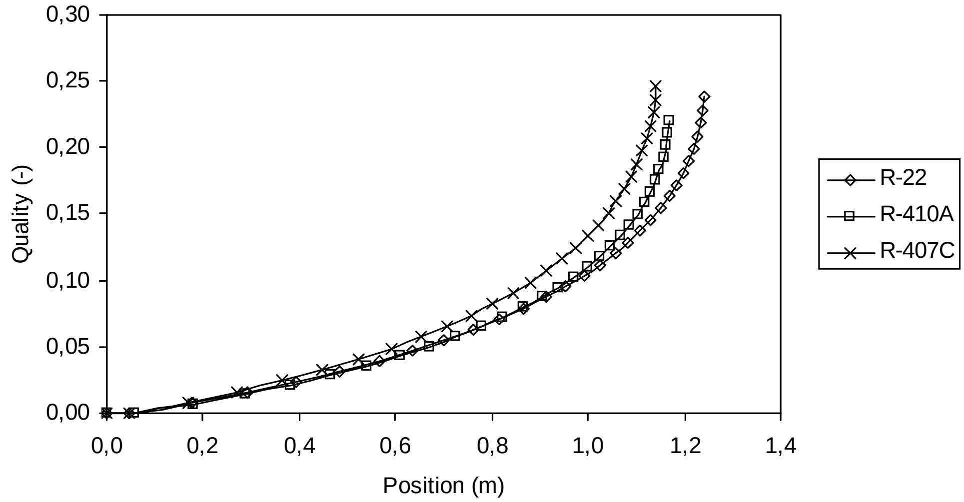

Figure 8: Quality profile along capillary tube

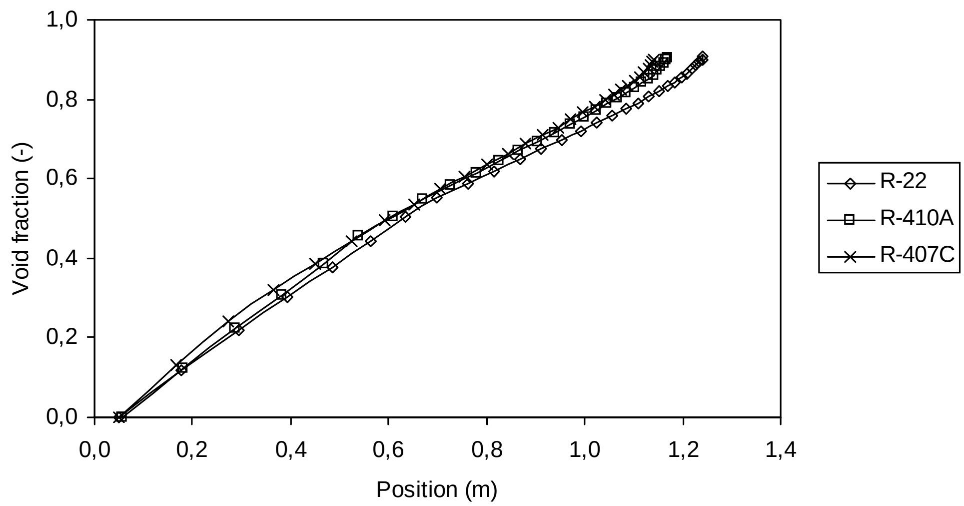

Figure 9: Void fraction profile along capillary tube

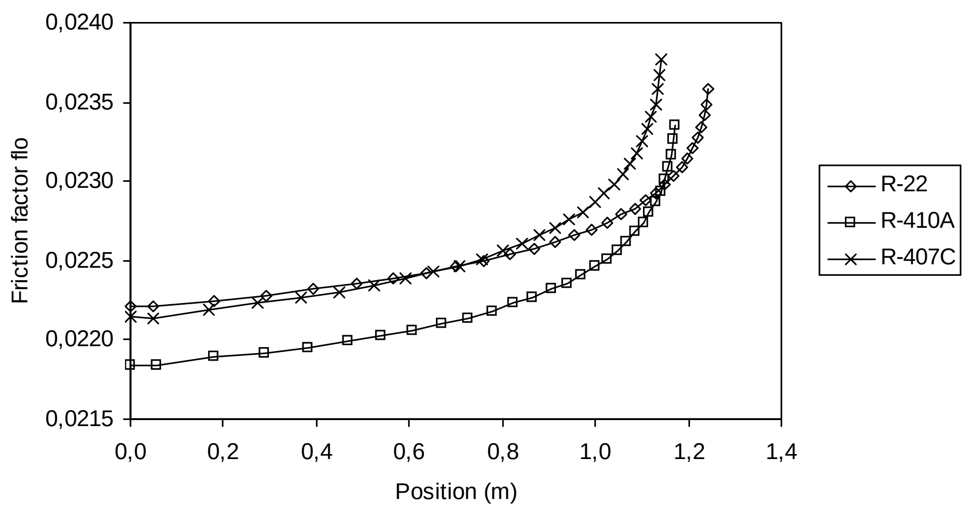

Figure 10. Single-phase friction factor profile along capillary tube

7. REFERENCES

1. ——, 1998, Equipment Available for Ozone-Safe Refrigerants. First Update - January 1998, *Bulletin of the International Institute of Refrigeration*, vol. LXXVIII n°. 2.
2. ——, 1997, *EES Engineering Equation Solver Software User's Guide*, F-Chart Software.
3. ——, 1996, *NIST Thermodynamic Properties of Refrigerants and Refrigerant Mixtures Database (REFPROP) - Version 5.0 - User Guide*, U.S. Dept. of Commerce, Gaithersburg.
4. Bittle, R.R.; Pate, M.B., 1996, A Theoretical Model for Predicting Adiabatic Capillary Tube Performance with Alternative Refrigerants, *ASHRAE Transactions*, vol. 102 part 2, pp.52-64.
5. Collier, J.G.; Thome, J.R., 1996, *Convective Boiling and Condensation*. 3rd ed. Oxford Science Publication, Oxford.
6. Fiorelli, F.A.S.; Peixoto, R.A.; Paiva, M.A.S.; Silvares, O.M., 1998, Numerical Study on Refrigerant Mixtures Flow in Capillary Tubes, *Proc. 1998 Int. Refrig. Conf. Purdue*, Purdue University, pp. 413-418
7. Heide, R., 1997, The surface tension of HFC refrigerants and mixtures. *Int. J. Refrig.*, vol. 20 no. 7, pp. 496-503.
8. Huerta, A.A.S.; Silvares, O.M., 1998, Simulation of the Effects of Oil in Capillary Tubes Considering a Separated Flow Model, *Proc. 1998 Int. Refrig. Conf. Purdue*, Purdue University, pp. 443-448
9. Idelcik, I.E., 1960, *Memento des Pertes de Charge*, Eyrolles Editeur, Paris.
10. Li, R.Y. et al., 1991, Numerical Analysis on Flow of Stratospherically Safe Refrigerant – R134a Through Capillary Tube, *Proc. Proc. 18th Int. Congr. Refrigeration*, IIR/IIF.
11. Pate, M.B., 1982, *A Theoretical and Experimental Analysis of Capillary Tube-Suction Line Heat Exchangers*. Ph.D. Thesis, Purdue University.
12. Paiva, M.A.S.; Fiorelli, F.A.S.; Peixoto, R.A.; Vodianitskaia, P.; Hernandez, A.N.; Silvares, O.M., 1995, Experimental and Numerical Study of the Flow Through Non Adiabatic Capillary Tubes with Lateral and Concentric Capillary Tube-Suction Line Heat Exchanger Configuration, *Proc. 19th Int. Congr. Refrig.*, IIR/IIF.
13. Peixoto, R.A., 1995, Experimental Analysis and Numerical Simulation of Capillary Tube-Suction Line Heat Exchangers Using Refrigerant HFC-134a, *Proc. 19th Int. Congr. Refrigeration*, IIR/IIF.
14. Peixoto, R.A.; Fiorelli, F.A.S.; Paiva, M.A.S.; Silvares, O.M., 1998, Numerical Simulation of Capillary Tube-Suction Line Heat Exchangers Using CFC-12, R-401A, R-409A and HC Blend, *Proc. Emerging Trends in Refrigeration and Air Conditioning (Joint Meeting of IIR Sections B/E)*, IIR/IIF.
15. Premoli, A.; Di Francesco, D.; Prina, A., 1971, Una Correlazione Adimensionale per la Determinazione della Densità di Miscele Bifasiche, *La Termotecnica*, vol. 25 no. 1, pp.17-25
16. Sami, S.M.; Tribes, C., 1998, Numerical Prediction of Capillary Tube Behaviour with Pure and Binary Alternative Refrigerants, *Applied Thermal Engineering*, vol. 18 n°.6, pp.491-502.
17. Sami, S.M.; Poirier, B.; Dahamani, A.B., 1998, Modelling of Capillary Tubes Behaviour with HCFC 22 Ternary Alternative Refrigerants, *Int. J. Energy Res.*, vol 22, pp.843-855.
18. Whalley, P.B., 1987, *Boiling, Condensation and Gas-Liquid Flow*, Oxford Science Publications, Oxford., p.68-77
19. Wong, T.N.; Ooi, K.T., 1996, Adiabatic Capillary Tube Expansion Devices: a Comparison of the Homogeneous Flow and Separated Flow Models, *Applied Thermal Engineering*, vol. 16 no. 7, pp.625-634