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NUMERICAL STUDY ON REFRIGERANT MIXTURES FLOW IN CAPILLARY TUBES

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ABSTRACT

This paper presents a numerical study on the performance of capillary tubes with R-22 and some alternatives considered for R-22 replacement, namely R-407C, R-410A and R-134a. It is compared the mass flow rate as function of inlet conditions (condensing temperature, 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%), R-410A presents about 30% higher mass flow rates and R-134a presents nearly 30% lower mass flow rates. These results are connected with the different saturation pressure curves for each fluid

NOMENCLATURE

A	Area	x	quality
c	sound speed	y	molar fraction
C_c	contraction coefficient	z	position
d	diameter	α	void fraction
f	Darci friction factor	ε_{rel}	relative roughness
G	mass flux	σ	area ratio
h	specific enthalpy		**subscripts**
h_c	convection heat transfer coefficient	cd	condenser, condensation
k	liquid conductivity	ct	capillary tube
$\dot{m}$	mass flow rate	ev	evaporator, evaporation
p	pressure	i	generic step
Pr	Prandtl number	in	inlet
Re	Reynolds number	l	liquid
T	temperature	o	ambient
UA'	external heat transfer coefficient	v	vapor
v	specific volume	w	wall

INTRODUCTION

In recent years it has been found evidences connecting halogenated compounds like refrigerants to the reduction of stratospheric ozone layer. These evidences led to the signature of the Montreal Protocol (1976), which main goal is the elimination of such substances. The R-12, used primarily in household refrigerators and freezers, is the most important and more dangerous refrigerant concerning to this deleterious effect. In developed countries the use of this fluid had already been eliminated since the beginning of 1996. Concerning to R-22, used mainly in air conditioning and heat pumps systems and less dangerous to ozone layer, a less restrictive schedule was established. Therefore, the priority of Refrigeration an Air Conditioning Industry was to find an alternative to CFC 12 and some pure substances, were selected as alternatives refrigerants in new refrigerators and freezers.

Attention is now addressed to R-22. This fluid is 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 great modifications in existing equipment. The HFC 134a is an alternative only for a narrow range of operational conditions. The use of non-azeotropic or near-azeotropic refrigerant mixtures are the most suitable alternative so far.

The use of such mixtures demands new experimental and numerical studies in order to evaluate how they affect the performance of refrigeration cycles as well as the design of their components. In this way the sizing of adiabatic capillary tubes (the expansion device used in small size air conditioners) using non-azeotropic mixtures is a subject of particular interest.

A recent survey (IIR, 1998) shows that compressors manufacturers as well as manufacturers of small and medium size air conditioning units are using mainly R-407C (a non-azeotropic 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 to small size units.

This paper presents the first results of a comparative study on the performance of HCFC 22 and some alternative refrigerant mixtures flowing through adiabatic capillary tubes (CT). The predicted performance of a capillary tube using R-22, R-407C, R-410A and R-134a are compared.

MATHEMATICAL MODEL

The mathematical model is based on previous works of the authors for non adiabatic tubes (Paiva et al., 1995; Peixoto, 1995; Peixoto et al., 1998). Some modifications were introduced in formerly models in order to calculate the properties of refrigerant mixtures.

The main assumptions of the 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 to/from ambient), no delay of vaporization, and homogeneous equilibrium two-phase flow model. According to Whalley (1996) the last assumption is a reasonable one for $G > 2000$ kg/s.m^2, which is true for air conditioners' CT.

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. Figure 1 shows the main variables and parameters involved in CT simulation. The governing equations are mass, momentum and energy balances, given by Eqs. (01) to (03).

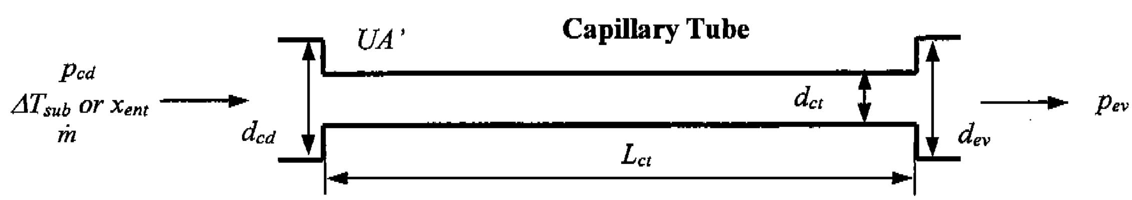

Figure 1. Variables of CT simulation model

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

$$(dp/dz) = -\left[\left(fvG^2\right)/\left(2d_{ct}\right)\right] - G^2\left(dv/dz\right) \tag{02}$$

$$\dot{m}\left(dh/dz\right) = -h_c \pi d_{ct}\left(T_{ct} - T_w\right) - \dot{m}vG^2\left(dv/dz\right) \tag{03}$$

Thermodynamic properties are calculated using REFPROP subroutines (NIST, 1996). Friction factor is calculated using Serghides correlation to Moody diagram (apud Kakaç et al., 1987), using Dukler correlation for calculation of homogeneous viscosity:

$$\left(1/\sqrt{f}\right) = A_5 - \frac{(A_5 - B_2)^2}{(A_5 + 2B_2 + C_1)} \tag{04}$$

where:

$$A_5 = -0{,}8686 \ln\left[(\varepsilon_{rel}/7{,}4) + (12/\mathrm{Re})\right]$$
$$B_2 = -0{,}8686 \ln\left[(\varepsilon_{rel}/7{,}4) + (2{,}51 A_5/\mathrm{Re})\right]$$
$$C_1 = -0{,}8686 \ln\left[(\varepsilon_{rel}/7{,}4) + (2{,}51 B_2/\mathrm{Re})\right]$$

Critical flow condition is verified by comparison of fluid velocity to sound speed at CT end, which is obtained by numerical differentiation of pressure with respect to density at constant entropy:

$$c^2 = \left.\left(\frac{\partial p}{\partial \rho}\right)\right|_{s=const} \tag{05}$$

Internal 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 (cf. Pate, 1982), Eq. (06), using the average velocity and liquid properties. Exponent n is 0,4 for heating or 0,3 for cooling.

$$(h_c d_{ct})/k_l = 0{,}023\,\mathrm{Re}_l^{0,8}\,\mathrm{Pr}_l^n\left[(1-x)/(1-\alpha)\right]^{0,8} \tag{06}$$

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. (07) and (08) respectively (cf. Collier & Thome, 1996), where $\sigma = A_{cd}/A_{ct}$ and $C_c = f(\sigma)$. For subcooled liquid inlet conditions pressure drop at inlet is calculated by Eq. (09) (cf. Idelcik, 1960).

$$p_{cd} - p_{in} = \frac{G^2 v_l}{2}\left[\left(\frac{1}{C_c} - 1\right)^2 + \left(1 - \frac{1}{\sigma^2}\right)\right]\left[1 + x(v_{lv}/v_l)\right] \tag{07}$$

$$p_{out} - p_{ev} = G^2 v_l \sigma (1-\sigma)\left[1 + x(v_{lv}/v_l)\right] \tag{08}$$

$$p_{cd} - p_{in} = 1{,}5\left[\left(G^2 v\right)/2\right] \tag{09}$$

CT wall temperature T_w is calculated by:

$$T_w = \frac{h_c \pi d_{ct} T_{ct} + UA' T_o}{h_c \pi d_{ct} + UA'} \tag{10}$$

From conservation equations it is obtained p and h 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-vapor thermal and hydrodynamic equilibrium, liquid and vapor composition along the CT is calculated. From composition, T and p it is obtained saturated liquid and vapor properties. At last, from h it is calculated x profile and mixture properties. Eqs. (11) to (16) show these calculations.

$$T_{ct,i} = f(p_i, h_i, y_1, ..., y_n) \quad (11)$$

$$y_{1,l,i}, ..., y_{n,l,i} = f(p_i, T_i, y_1, ..., y_n) \quad (12)$$

$$y_{1,v,i}, ..., y_{n,v,i} = f(p_i, T_i, y_1, ..., y_n) \quad (13)$$

$$h_{l,i} = f(p_i, x = 0, y_{1,l}, ..., y_{n,l}) \quad (14)$$

$$h_{v,i} = f(p_i, x = 1, y_{1,v}, ..., y_{n,v}) \quad (15)$$

$$x_i = (h_i - h_{l,i})/(h_{v,i} - h_{l,i}) \quad (16)$$

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 nonlinear algebraic equations using a modified Newton-Raphson method. The step variable used is the pressure drop.

The experimental validation of model for single refrigerants simulation (CFC-12 and HFC-134a) was presented in previous works (Peixoto, 1995; Paiva, Fiorelli et al., 1995). It was also compared simulation results with experimental data from literature and from experimental tests performed (Dirik et al., 1994; Escanes, 1995; Whitesel, 1957; Kuehl & Goldshmidt, 1990). The calculated mass flow rates agree with experimental and literature data within 10%.

RESULTS AND CONCLUSION

Calculations were performed for the following geometry: d_{ct} = 1.676 mm (0.066"), L_{ct} = 1.524 m (60"), $d_{cd} = d_{ev}$ = 6 mm (0.236"). It was assumed constant ambient temperature T_o = 25°C (77°F) and UA' = 0.11 W/m.°C (53 Btu/h.in.°F) as well as a constant evaporation temperature T_{ev} = -5°C (23°F).

Figs. 2 to 5 show the numerical results obtained. Taking R-22 as the reference, R-407C mass flow rates are slightly higher (about 5%) than R-22, R-410A flow rates are about 30% higher, and R-134a flow rates are nearly 30% lower.

These results are primarily connected with different saturation pressure curves for each fluid, as shown in Fig. 6. From these results it is reasonable to consider R-407C as a good "drop-in" alternative when the mass flow rate is taken as comparison parameter. R-410A and R-134a should only be used in new equipment suitably redesigned. It is important to point out that only mass flow rate has been analyzed in this paper. There are other aspects to be considered. For instance, Peixoto et al. (1998), compared both mass flow rate and cooling capacity for some "drop-in" alternatives to R-12.

Concluding, obtained results agrees with literature, which indicates R-407C as a drop-in refrigerant as well as to new equipment, and the two others (R-410A and R-134a) only to new equipment. These are the first results of a research project in development at University of São Paulo. Next step on this project is to obtain experimental data of non-azeotropic mixtures flowing through adiabatic capillary tubes in order to validate the simulation model briefly described in this paper.

ACKNOWLEDGEMENTS

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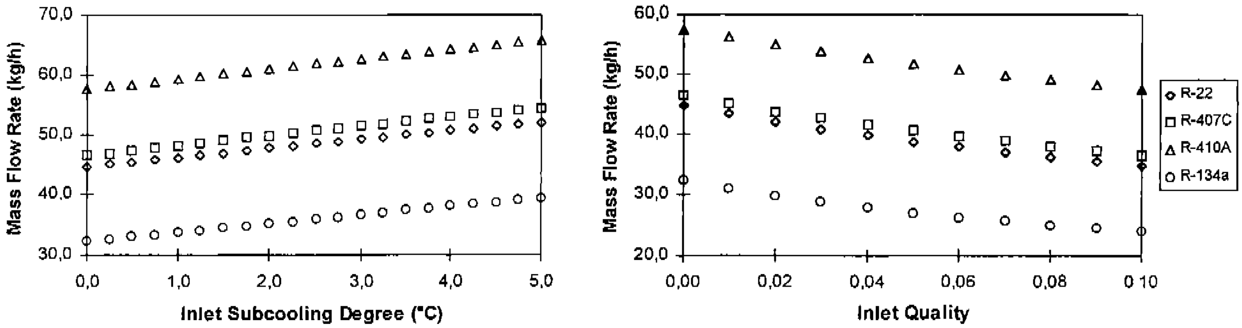

Figure 2. Mass flow rate as function of x_{in} and $\Delta T_{sub,in}$ for 40°C (104°F) condensing temperature

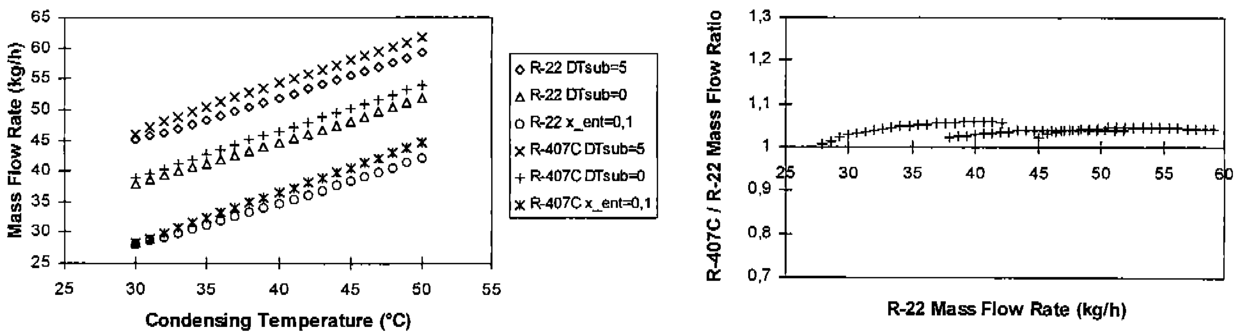

Figure 3. Comparison of mass flow rate as function of T_{cd}, x_{in} and $\Delta T_{sub,in}$ for R-22 and R-407C

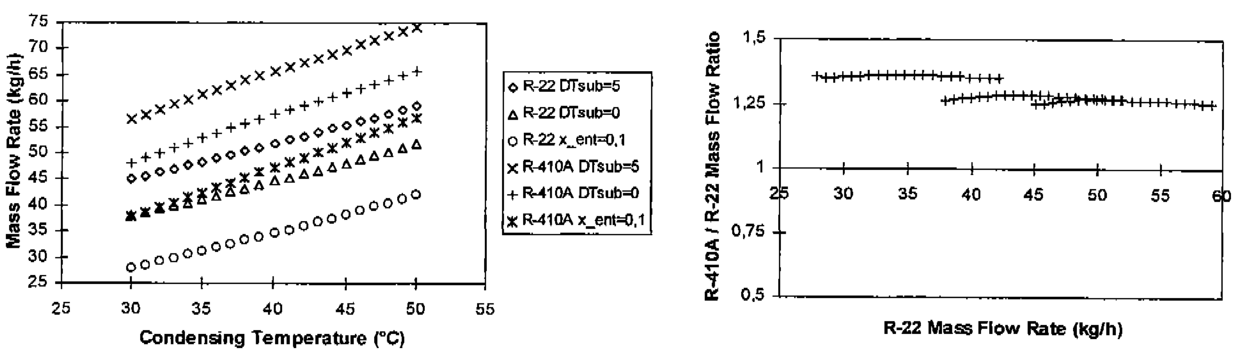

Figure 4. Comparison of mass flow rate as function of T_{cd}, x_{in} and $\Delta T_{sub,in}$ for R-22 and R-410A

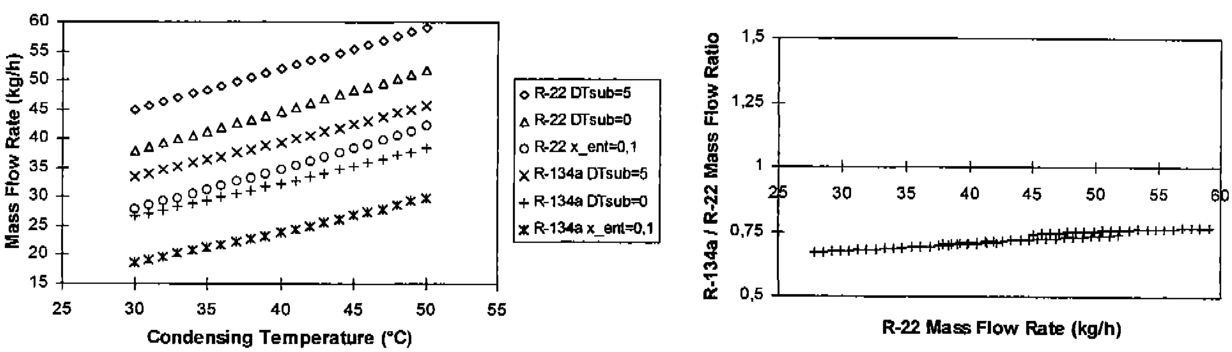

Figure 5. Comparison of mass flow rate as function of T_{cd}, x_{in} and $\Delta T_{sub,in}$ for R-22 and R-134a

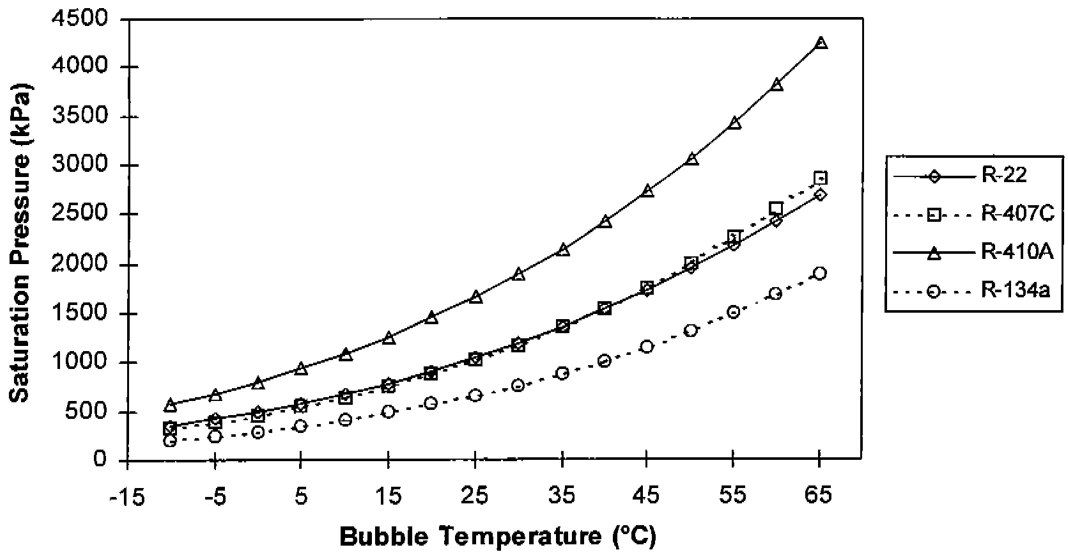

Figure 6. Saturation pressure as function of bubble temperature for R-22, R-407A, R-410A and R-134a

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