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EXPERIMENTAL VALIDATION OF A CAPPILARY TUBE SIMULATION MODEL WITH REFRIGERANT MIXTURES FLOW

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Abstract. *This paper presents the experimental validation of a simulation model for refrigerant mixtures flow through capillary tubes. To perform such validation it was built an experimental apparatus using a blow-down process. It was carried out preliminary tests for characterization of experimental parameters: actual capillary tube diameters; relative roughness; and the heat losses in subcooling/quality control system. It was obtained almost 200 experimental points 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 delay of vaporization phenomenon occurrence, experimentally verified by the authors.*

Keywords: *capillary tubes, refrigerant mixtures flow, modelling, simulation*

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 great modifications in existing equipment.

The use of zeotropic or near-azeotropic refrigerant mixtures is 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 cycle component. In this way the sizing of adiabatic capillary tubes using zeotropic mixtures is a subject of particular interest. It can be pointed out some works in this area in recent years (Bittle & Pate, 1996; Chang & Ro, 1996; Bittle et al., 1998; Sami & Tribes, 1998; Sami et al., 1998; Fiorelli et al., 1998; 1999; Wei et al., 1999; Jung et al., 1999; Motta et al., 2000; Fiorelli, 2000).

This paper presents the experimental validation of a simulation model for refrigerant mixtures flow through capillary tubes (Fiorelli et al., 1998). To perform such validation it was built an experimental apparatus using a blow-down process. It was carried out preliminary tests for characterization of some experimental parameters: actual capillary tube diameters by mercury filling a tube sample; relative roughness by measurement of pressure drop of an all-liquid flow; and the heat losses in subcooling/quality control system by performing an energy balance for such subsystem.

It was obtained almost 200 experimental points 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.

2. Mathematical Model

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 capillary tube walls, constant external UA' coefficient, 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.m2, which usually is true for capillary tubes.

For zeotropic 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 capillary tubes simulation. The governing equations are mass, momentum and energy balances, given by Eqs. (1) to (3).

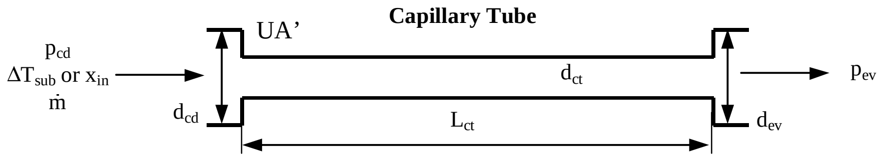


Figure 1. Variables of capillary tubes simulation model

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

$$dp/dz = -fvG^2/(2d_{ct}) - G^2\, dv/dz \tag{2}$$

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

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:

$$\frac{1}{\sqrt{f}} = A_5 - \frac{(A_5 - B_2)^2}{(A_5 + 2B_2 + C_1)} \tag{4}$$

where

$$A_5 = -0.8686 \ln(\varepsilon_{rel}/7.4 + 12/Re) \tag{5}$$

$$B_2 = -0.8686 \ln(\varepsilon_{rel}/7.4 + 2.51A_5/Re) \tag{6}$$

$$C_1 = -0.8686 \ln(\varepsilon_{rel}/7.4 + 2.51B_2/Re) \tag{7}$$

Critical flow condition is verified by comparison of fluid velocity to sound speed at capillary tube end, given by:

$$c^2 = (\partial p/\partial \rho)_{s=const} \tag{8}$$

Internal heat transfer coefficient hc 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. (9), using the average velocity and liquid properties. Exponent n is 0.4 for heating or 0.3 for cooling.

$$h_c d_{ct}/k_l = 0.023 Re_l^{0.8} Pr_l^n \left[(1-x)/(1-\alpha)\right]^{0.8} \tag{9}$$

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. (10) and (11) 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. (12) (cf. Idelcik, 1960).

$$p_{cd} - p_{in} = \frac{G^2 v_l}{2}\left[\left(\frac{1}{C_c} - 1\right)^2 + \left(1 - \frac{1}{\sigma^2}\right)\right]\left(1 + x\frac{v_{lv}}{v_l}\right) \tag{10}$$

$$p_{out} - p_{ev} = G^2 v_l \sigma (1-\sigma)(1 + x\, v_{lv}/v_l) \tag{11}$$

$$p_{cd} - p_{in} = 0.75 G^2 v \tag{12}$$

Capillary tube wall temperature T_w is calculated by:

$$T_w = (h_c \pi d_{ct} T_{ct} + UA' T_o)/(h_c \pi d_{ct} + UA') \tag{13}$$

From conservation equations it is obtained p and h along the capillary tube. 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 capillary tube 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. Equations (14) to (19) show these calculations.

$$T_{ct,i} = f(p_i, h_i, y_1, ..., y_n) \tag{14}$$

$$y_{1,l,i}, ..., y_{n,l,i} = f(p_i, T_i, y_1, ..., y_n) \tag{15}$$

$$y_{1,v,i}, ..., y_{n,v,i} = f(p_i, T_i, y_1, ..., y_n) \tag{16}$$

$$h_{l,i} = f(p_i, x = 0, y_{1,l}, ..., y_{n,l}) \tag{17}$$

$$h_{v,i} = f(p_i, x = 1, y_{1,v}, ..., y_{n,v}) \tag{18}$$

$$x_i = \left(h_i - h_{l,i}\right) / \left(h_{v,i} - h_{l,i}\right) \tag{19}$$

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

3. Experimental Apparatus

Figure (2) presents a flowchart of the experimental apparatus built to this study. It is used a blow-down process in order to provide an accurate control of the process parameters. Refrigerant is initially stored upward the test section in a high-pressure reservoir (two bladder accumulators of 100 litres each). Such high pressure is provided by nitrogen filling of the bladders. Test section exit is connected to a low-pressure reservoir. Low pressure is obtained by refrigerant condensation provided by chilled ethylene-glycol/water mixture flowing through a coil inside the reservoir. By pressure difference the refrigerant flows from the high-pressure to the low-pressure reservoir through the test section where is placed the capillary tube (CT). Fluid return is also made by pressure difference. Pressure is raised in the low-pressure reservoir by means of two heaters, as long as pressure is lowered in the high-pressure reservoir by releasing nitrogen.

Parameters measured in test section are the pressure and temperature profiles along CT, mass flow rate through CT, electric power consumption of subcooling/quality final control heater, and mixture composition at CT inlet. 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).

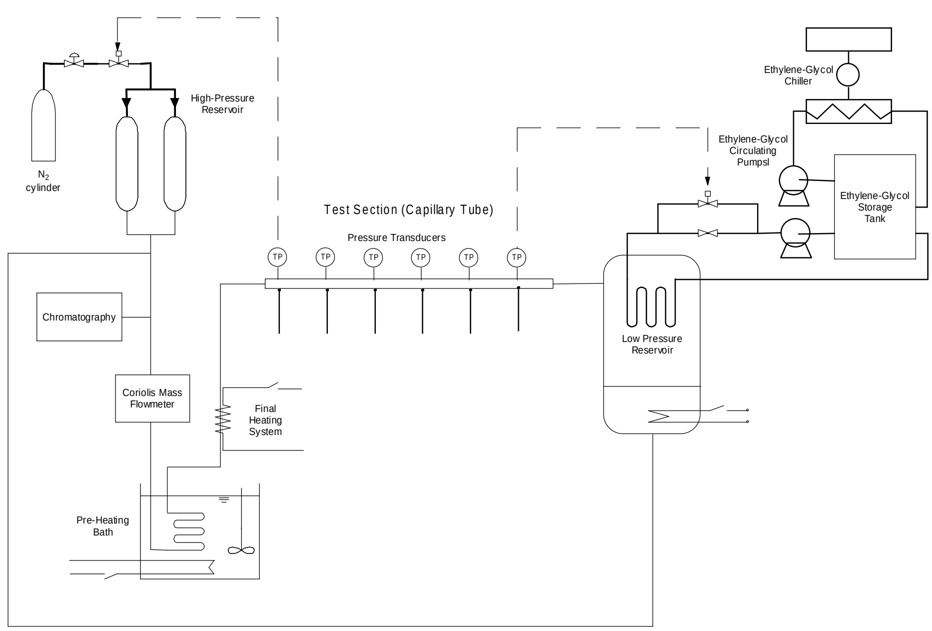


Figure 2. Experimental apparatus flowchart

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.

The main parameters to be controlled in each run are the CT inlet and outlet pressures, and CT inlet subcooling or quality. CT Inlet pressure is controlled by a regulator (pre-control) and a PID control/solenoid valve system (final control) on N_2 line. Outlet pressure is controlled by a PID control/solenoid valve system on ethylene-glycol line. By controlling the glycol flow rate, it is possible to control condenser pressure and, as a result, the CT outlet pressure. At last, subcooling/quality control is achieved by a heating system divided in two parts. It is performed a pre-heating by a coil immersed in a hot water constant-temperature bath. Final adjustment of subcooling or quality is obtained by a 5.0 W/m tape heater and a PID controller (subcooling) or manual control (quality).

4. Experimental Analysis

First step in obtaining the experimental data was the characterisation of some experimental parameters: actual capillary tube diameters and relative roughness, the heat losses in subcooling/quality control system, as well as an evaluation of instrumentation effect on mass flow rate, particularly the pressure taps.

Actual CT internal diameter was measured by filling a CT sample with mercury. This procedure provides the average diameter with ± 1.0% uncertainty. CT diameters used in this work were 1.101 mm, 1.394 mm and 1.641 mm.

Relative roughness was evaluated by measuring the pressure losses of an all-liquid R-410A refrigerant flow through CT. From these pressure losses it was possible to calculate the friction factors. Using the Colebrook equation and the calculated friction factors, relative roughness was evaluated. Obtained values for the above listed diameters were 2.354×10^{-4}, 3.604×10^{-4}, and 2.193×10^{-4} respectively. This value is smaller than others find in open literature, and is due to manufacturing process, that includes a polishing phase.

Evaluation of heat losses in subcooling/quality control system was performed in order to determine how much of the electric power provided to the tape resistance is in fact used to raise refrigerant enthalpy. Heat losses were evaluated by measurements of inlet and outlet temperatures, as well as mass flow rate and resistance electric power consumption of an all-liquid refrigerant flow through the heat exchanger. These results were used for evaluation of capillary tube inlet quality . The uncertainty of such evaluation was of about ± 10%.

Finally, it was evaluated the effect of instrumentation on capillary tube performance. Such instrumentation, particularly the pressure taps, may significantly reduce refrigerant mass flow rate through the capillary tube due to the introduction of an additional local pressure loss or the shift of flashing point, since the pressure taps may act as active cavities for bubble nucleation. According to literature (cf. Meyer & Dunn, 1998), the reduction in mass flow rate may be up to 15%. Another factor that can affect capillary tube behaviour is thermocouples soldering on tube wall. This would locally change heat transfer, once thermocouples act as fins and may consequently create conditions for appearance of active cavities where flashing inception takes place. Such evaluation was performed by comparing a set of 54 experimental, points obtained for a given capillary tube without instrumentation, with a second data set obtained for the same operational conditions after the instrumentation were attached to the capillary tube. Such comparison showed a small effect of instrumentation on capillary tube performance (about 1.5% mass flow rate average reduction).

It was obtained three experimental points at each run. The experimental procedure consisted in fixing inlet and outlet pressure, as well as the initial subcooling or quality, and then starts the test/data acquisition. Once the first steady state operational condition is reached, it is annotated time and recorded a data set. Then a new subcooling or quality value is fixed, and again we wait until a new steady state operational condition is reached to annotate time and record data. The same steps are performed to the third point. In parallel, it is performed three to five chromatographic analyses in order to evaluate refrigerant composition.

Steady state verification criterion is the elapsed time since last parameters setting. It was observed during preliminary tests that it was necessary about 50 minutes to reach the first steady state condition. Next operational conditions took close to 20 minutes to steady state. Thus it was adopted standard intervals of respectively one hour and 30 minutes for reaching steady state conditions. Such criterion was adopted in parallel to operator’s monitorship during the tests. Later data analysis showed that the adopted standard intervals were suitable to all performed tests.

Once the experimental apparatus was mounted and adjusted, it was performed the data survey for the following refrigerant mixtures: R-410A (50% R-32/50% R-125 on mass basis) and R-407C (23% R-32/25% R-125/52% R-134a).

For R-410A it was performed 28 runs, totalling 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, totalling 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. A table set containing all experimental data is about to be published (Fiorelli et al., 2002).

Figure (3) shows typical experimental temperature profiles for subcooled and two-phase inlet conditions. Pressure profiles are presented in Figs. (4) and (5) for both refrigerant mixtures. In this figures it is also shown the saturation pressure profile evaluated for the given temperature profile and refrigerant composition, assuming zero quality.

These figures also show that temperature, pressure and composition measurements are consistent, once there is a good agreement between measured and calculated pressure profiles for R-410A. By the other hand, for R-407C it can be seen in fig. 5b that there is a detachment between the measured pressure profile and the calculated (bubble pressure) one as the quality grows. This figure also shows that the measured profile approaches the calculated dew pressure profile. Such behaviour is expected once a zeotropic mixture presents composition variation during phase change process, which leads to a saturation temperature glide (as well as a pressure glide) associated to this composition variation. In order to get agreement between measured and calculated profiles it is required a quality or composition profile along the flow. It can also be noticed in Figs. (4) and (5) the occurrence of the delay of vaporisation, detailed on Fig. (6). Such phenomenon occurred for most of the subcooled inlet runs for both fluids.

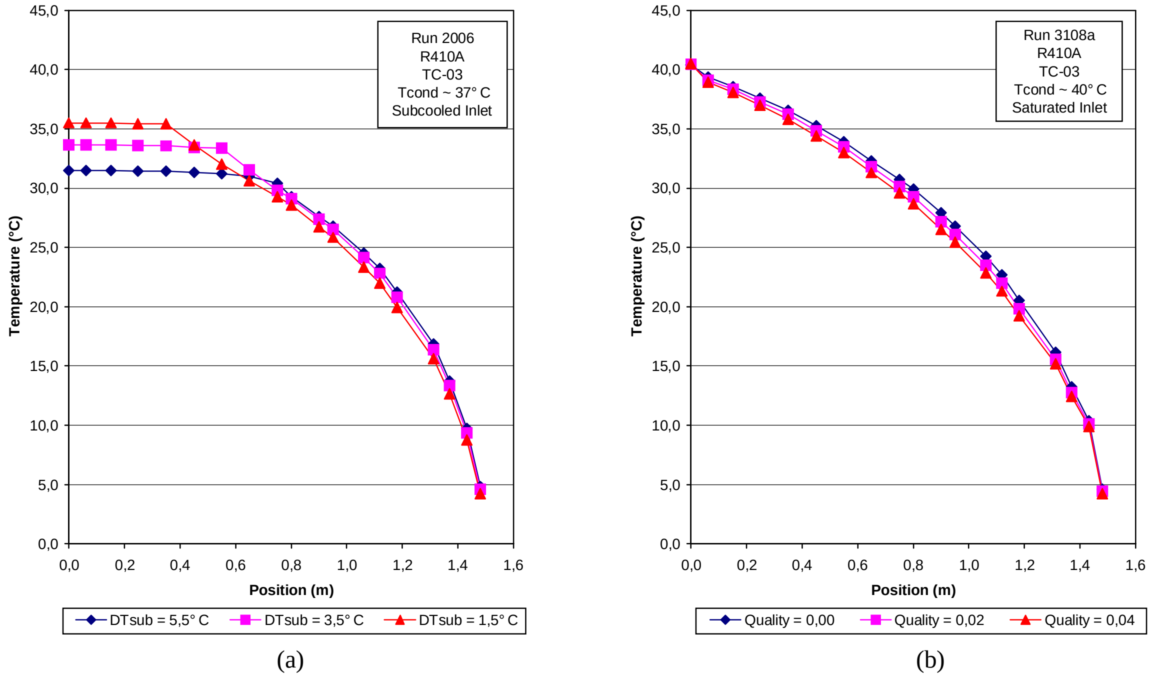


Figure 3. Typical temperature profiles for (a) subcooled and (b) two-phase inlet conditions

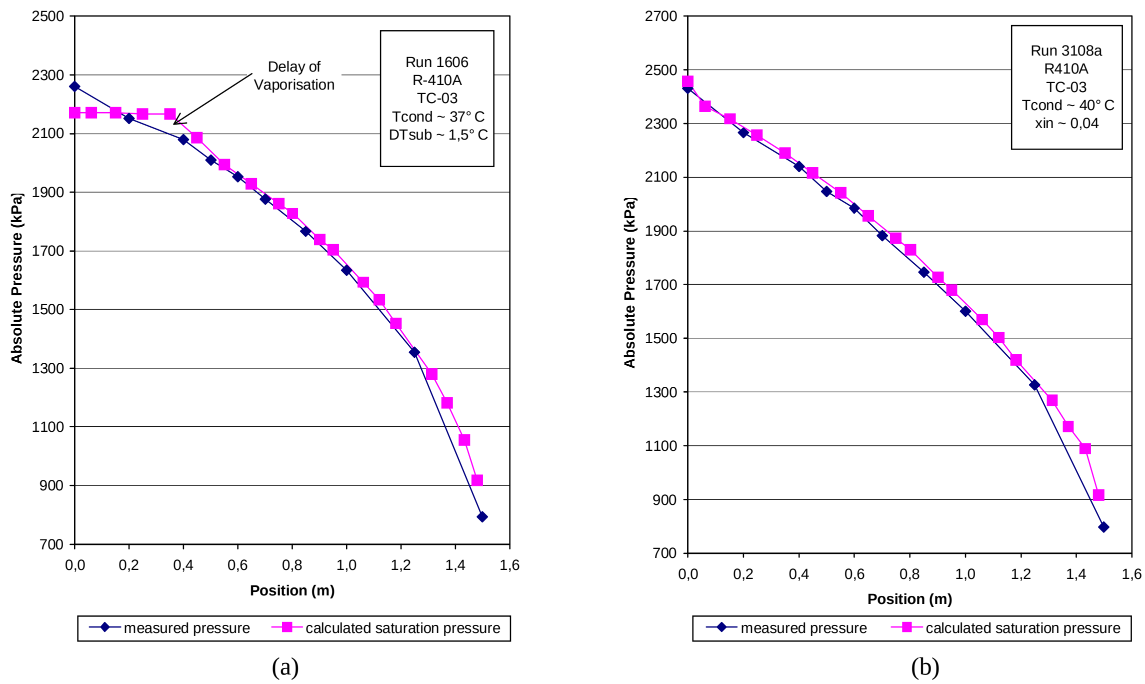


Figure 4. Pressure profiles for R-410A - (a) subcooled and (b) two-phase inlet conditions

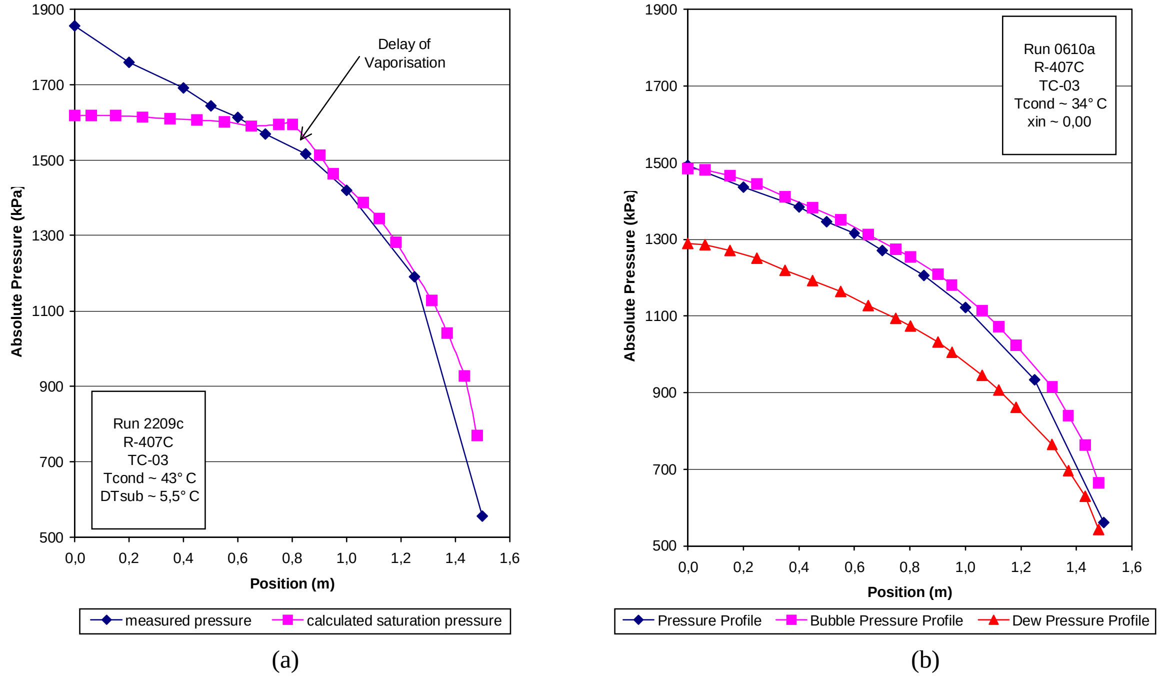


Figure 5. Pressure profiles for R-407C - (a) subcooled and (b) two-phase inlet conditions

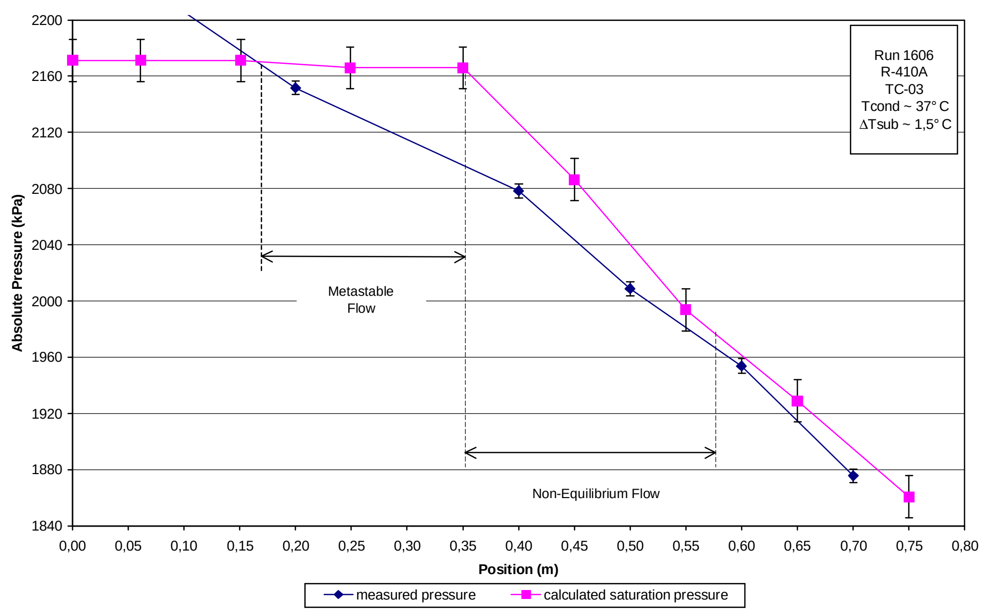


Figure 6. Detail of pressure profile showing the delay of vaporisation occurrence

5. Model Validation

5.1. R-410A

Figure (7) the comparison between calculated and experimental mass flow rate values for R-410A for both subcooled and two-phase inlet condition. It can be seen that for subcooled inlet conditions the simulation model underpredicts mass flow rate with an average deviation of -4.0% and a dispersion of ± 2.5%. For two-phase inlet it is obtained the same behaviour (mass flow rate underprediction), but deviation are greater (an average deviation of -5.8% and dispersion of ± 3.0%.

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.

Figure (8) presents the comparison of measured and calculated pressure and 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 vaporisation occurs. This phenomenon, detailed on Fig. 6, is one of the causes that lead to the mass flow rate underestimation. The smaller errors 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 vaporisation could improve overall capillary tube simulation model. For example, by adding the Δp_{sat} values obtained from experimental data to the inlet pressure provided to simulation model it is obtained an average error reduction from -4.0% to -1.5%.

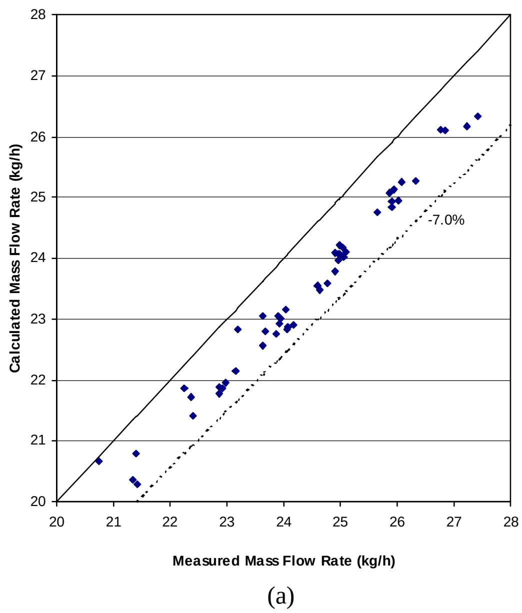


Figure 7. Comparison between measured and calculated mass flow rate for R-410A: (a) subcooled and (b) two-phase inlet conditions

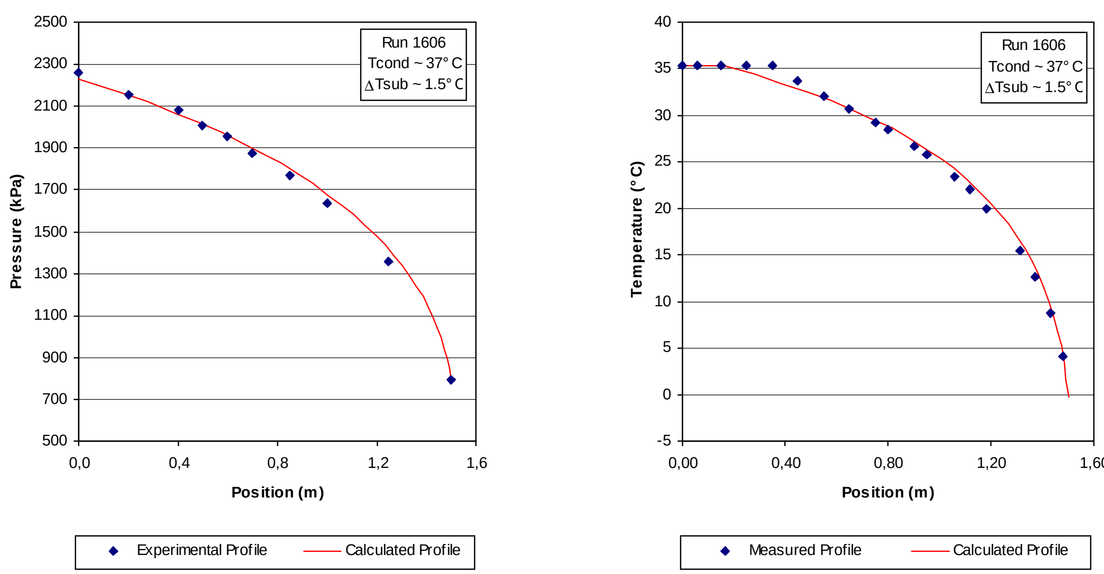


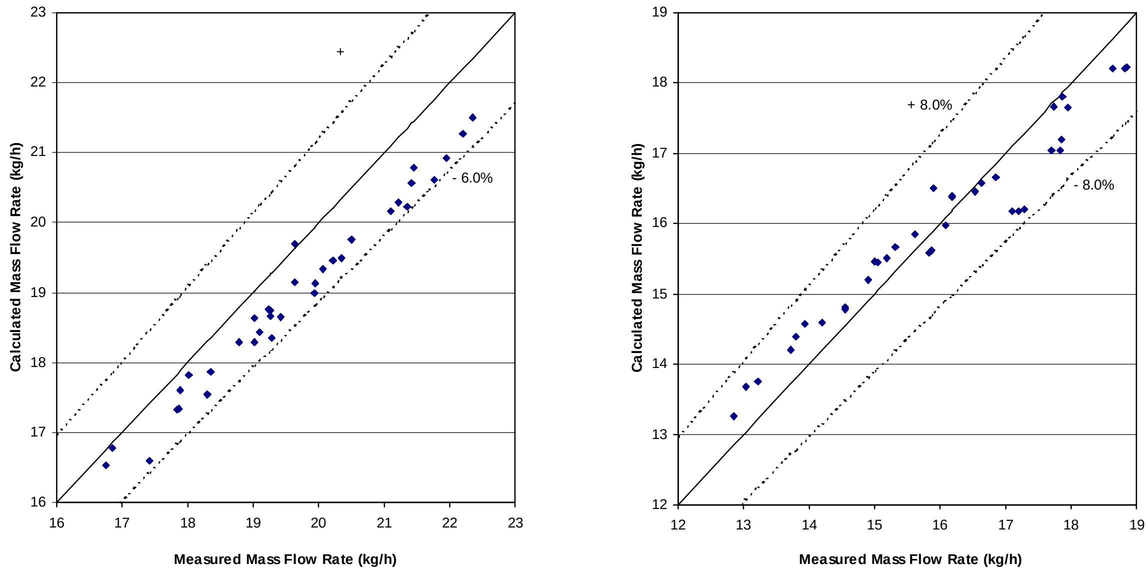
Figure 8. Pressure and temperature profiles for R-410A: subcooled inlet conditions

5.2. R-407C

Figures (9) and (10) show the comparison between experimental and calculated data for R-407C. For subcooled inlet, the model presents a similar behaviour to that for R-410A: it underestimates mass flow rate with an average error of -4.0% and a dispersion of ± 2.5%. However, deviations do not present an increasing tendency as subcooling decreases.

For two-phase inlet conditions, the model presents an average error of +0.1%. In this case, models behaviour 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 vaporisation occurs. Introduction of experimental values for Δp_{sat} changes average deviation from -4.0% to +1.5%.



(a) (b)

Figure 9. Comparison between measured and calculated mass flow rate for R-407C: (a) subcooled and (b) two-phase inlet conditions.

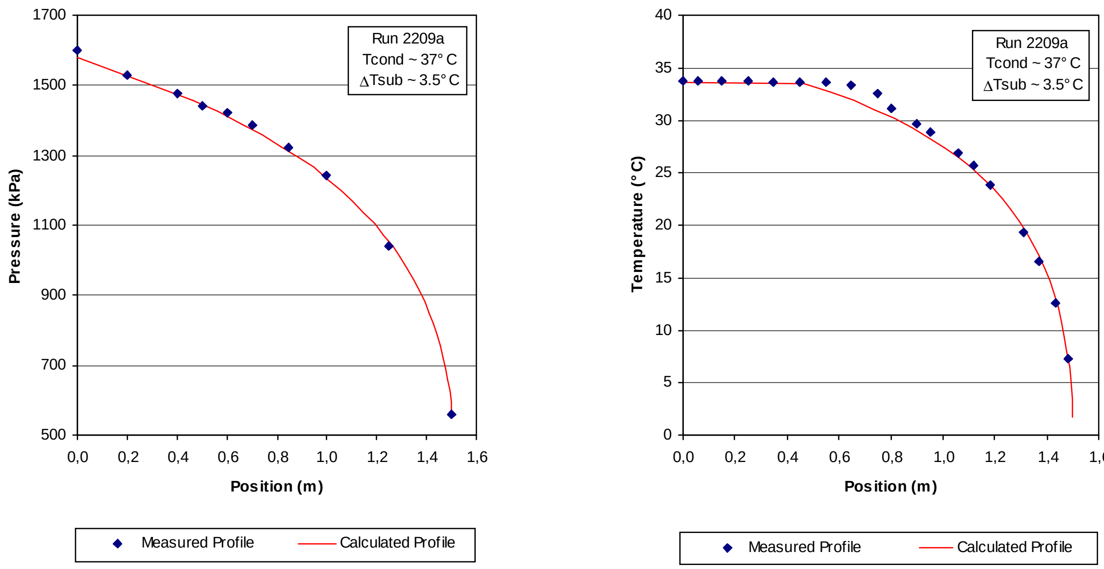


Figure 10. Pressure and Temperature Profiles for R-407C - Subcooled Inlet Conditions

6. Conclusion

This work presents the a simulation model for refrigerant mixtures flow through adiabatic capillary tubes. It is discussed the main assumptions for the model, and it is presented the governing and constitutive equations.

Experimental validation shows the model presents the same error level when compared to measured values of mass flow rate, temperature and pressure profiles for both R-410A and R-407C. The validation also shows the occurrence of the delay of vaporisation, which leads to a mass flow rate underestimation tendency. Taking account this phenomenon reduced error level for both fluids.

The presented results show that the model is suitable for simulation of adiabatic capillary tubes with refrigerant mixtures flow.

7. Acknowledgements

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