2004

# Experimental Study on Adiabatic Flow of R-22 Alternatives in Capillary Tubes

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# EXPERIMENTAL STUDY ON ADIABATIC FLOW OF R-22 ALTERNATIVES IN CAPILLARY TUBES

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## ABSTRACT

This paper presents an experimental research effort on the flow of R-22 and its alternatives in adiabatic capillary tubes. An experimental apparatus was developed and used to generate a matrix of experimental data, which was then used to develop empirical correlations to estimate the refrigerant mass flow rate as a function of the operating conditions (degree of subcooling and inlet pressure) and capillary tube geometry (internal diameter and length). The experiments were planned and performed following a statistically based methodology and specifically for the range of commercial refrigeration. Four refrigerants were considered, namely R-22, R-507a, R-404a and R-407c. The correlations obtained from the database yielded maximum relative deviations between 3.2 and 7.4%. An assessment of the ASHRAE's generalized correlation was also made by comparing its predictions with the measured data. It was found that the ASHRAE's correlation predicted the experimental data with a maximum relative deviation of 20%.

## 1. INTRODUCTION

A capillary tube is a constant area expansion device which has been widely used in small and medium sized refrigeration systems. Among its main advantages are the inexistence of movable parts, the low cost and the requirement for compressors of low starting torque.

One of the major disadvantages in adopting a capillary tube as an expansion device is its inability to respond to the variations of the system load conditions. Refrigerant systems employing capillary tubes have, therefore, a unique optimum operation point. Deviations from this point are always accompanied by a decrease of the system coefficient of performance. The selection of the appropriate capillary tube for a given application is, therefore, closely related to the refrigeration system performance.

A capillary tube appears quite simple, but the refrigerant flow inside this component is very complex. The refrigerant leaves the condenser and reaches the capillary tube usually as a saturated or subcooled liquid. At the tube entrance there is a slight pressure drop, due mainly to the abrupt change in cross sectional area. The pressure then decreases linearly along the tube length due to wall friction. This type of regime holds until the flow reaches saturated conditions. From this point (flash point), where vapor first appears, the pressure drop per unit length increases as the end of the tube is approached. This happens because the pressure drop in the two-phase flow region corresponds to both flow friction and the increase in momentum of the flow media.

Another important phenomenon in capillary tube flow is the metastability, characterized by the persistence of the liquid state to pressures less than the saturation pressure corresponding to its temperature. This causes a delay in the inception of vaporization, increasing the liquid length in the capillary tube and thus the refrigerant mass flow rate.

The refrigerant flow may also become choked at the capillary tube exit, when the fluid velocity reaches the local sonic velocity. After that any further lowering of the evaporating pressure has little effect on the mass flow rate (Stoecker, 1982).

Inumerous scientific projects have been developed in the last 50 years with the intention of exploring the refrigerant flow through capillary tubes and establishing appropriate rating procedures. More recently, several researchers have focused their attention on the study of capillary flow with refrigerant mixtures (Kim et al., 2002, Choi et al., 2003, Choi et al., 2004).

Mixtures of refrigerants are solutions of two or more components. Figure 1 displays a typical phase diagram for a zeotropic mixture, like R-407c. The main characteristic of this type of mixture is the temperature variation during the process of phase change at constant pressure.

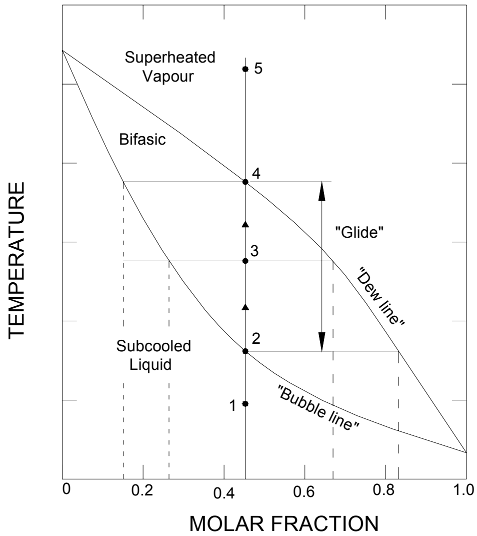

Figure 1: Phase diagram

Mixtures of some refrigerant fluids and in certain concentrations, however, behave as pure substances. In other words, they have a single saturation temperature for each pressure. These are called azeotropic mixtures, a notable example of which is R-507a. Mixtures of refrigerant fluids, whose behaviors are close to, but not exactly, that of azeotropic mixtures, are denominated near azeotropic mixtures, such as R-404a. These mixtures have a small temperature glide and, like the zeotropic mixtures, different concentrations in the liquid and vapor phases.

## 2. EXPERIMENTAL APPARATUS

The capillary tube test facility was a conventional vapor compression refrigeration loop assembled with a series of devices that allowed a controlled utilization within a broad range of operating conditions. A schematic diagram of the experimental apparatus is shown in Figure 2.

The condensation pressure was controlled by the heat released in the condenser. The condenser was a tube-in-tube water heat exchanger. The water flow was established by a pressure regulating valve (PCV), connected to a pressure tap installed between the coalescent filters. The condenser was thermally insulated with ceramic wool in order to reduce the condensation pressure variation with the room temperature. The room temperature during all the experiments was maintained at 21 ± 3ºC.

A water heat exchanger, also of the tube-in-tube type, was used to increase the refrigerant subcooling at the condenser exit. Four stainless steel electric heaters (HSB) were used to fine tune the refrigerant subcooling at the capillary tube inlet. The maximum power supplied by each heater was 800W at 220V. The heaters were assembled in pairs and in-series, and insulated with ceramic wool. The power released by the heaters was controlled by a PID system.

Four oil separators (OS1-1, OS1-2, OS2-1, OS2-2) were installed in the discharge line and connected to the process tubes of the compressors (C1, C2). The oil separators were heated by electric heaters (HOS1-1, HOS1-2, HOS2-1, HOS2-2) to facilitate the process of oil separation from the refrigerant. Two coalescent oil filters (CF1, CF2) were also installed in the discharge line. The efficiencies of the filters CF1 and CF2 were 99.97% and 99.999%, respectively.

Several valves (V1 to V9 and S1 to S8) were strategically installed in the experimental apparatus to facilitate its maintenance. The shut-off valves (S1 to S8) were mainly used for charging and evacuation of the system. The V17 and V19 valves were used to set the zero of the mass flow meter. The V18 is a by-pass valve for the mass flow rate measurement system.

The evaporator (HE) was an air heat exchanger. The air was circulated by two fans installed in-series (FAN E and FAN HA) and heated by two sets of electric heaters (HE and HA). The HE set consisted of two 400W power heaters, while the HA set consisted of five heaters with power varying between 200W and 800W.

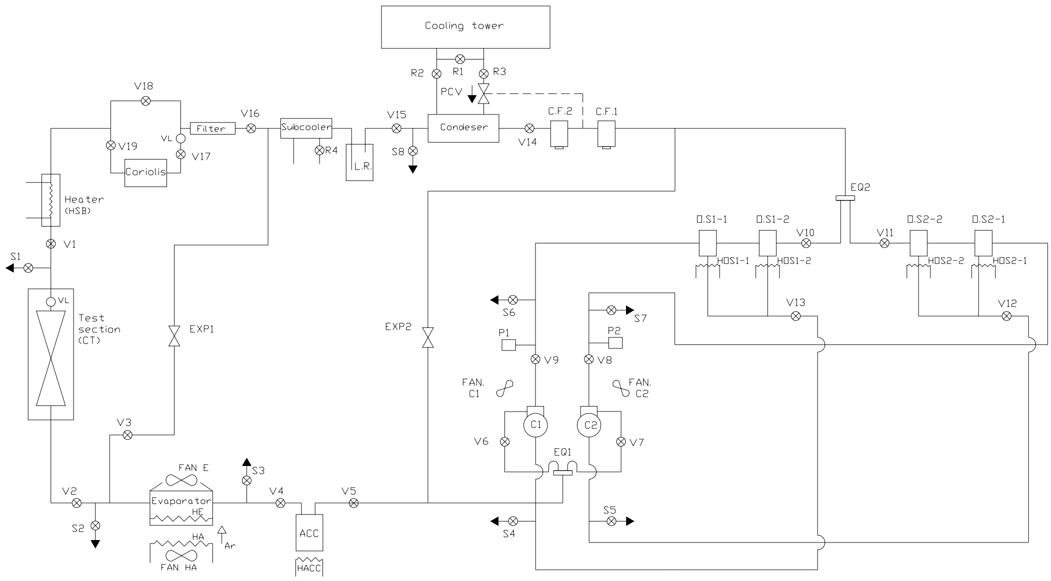


Figure 2: Schematic diagram of the experimental apparatus

## 3. TEST SECTION

The capillary tube was placed inside a partially dismountable wooden box, filled with styropor blocks that guaranteed the necessary thermal insulation. A section of (¼" O.D.) tubing was soldered at both extremities of the capillary tube, to facilitate its replacement. Two tubes of (½" O.D.) were fastened onto rigid bases to maintain the capillary tube as straight as possible. Figure 3 displays a schematic diagram of the test section.

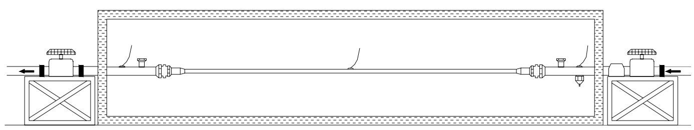

Figure 3: Test section

The pressure measurements were taken by strain gauge pressure transducers connected by small pieces of capillary tubes to 2mm holes made in the ½" piping. Pressure transducers of 50 and 20 bar, with a maximum uncertainty of ± 0.01 bar, were used to measure the capillary tube inlet and exit pressures, respectively.

The refrigerant temperature at the inlet of the capillary tube was measured by a T-type thermocouple, 0.13 mm in diameter and with a maximum uncertainty of ± 0.2°C. Thermocouples were also placed along the length of the capillary tube to determine the temperature distribution and were attached to the outside surface of the capillary tube with silver tape and thermal paste. Heat losses from the thermocouples were minimized by rolling them around the tube.

The refrigerant mass flow rate was measured by a Coriolis type mass flow meter, composed of a sensor and an electronic unit. The sensor was set up on a rigid base and connected to the refrigeration loop by flexible hoses to avoid eventual vibrations coming from the test facility. An interface was also used to adjust the measurement range of the transducer according to the test being performed. This procedure guaranteed a maximum uncertainty of ± 0.28 kg/h.

## 4. MEASUREMENT OF THE INNER DIAMETER OF THE CAPILLARY TUBES

Figure 4: Capillary tube sample

The measurement of the inner diameter of the capillary tubes is crucial for this kind of study, although unfortunately several researchers don't do this with the due care. The measurement process consisted of the preparation of samples of capillary tubes using pieces of epoxy resin, as illustrated in Figure 4.
The tube pieces were filled with resin to avoid deformations during the sample sanding and polishing processes. The perpendicularity of the tubes in relation to the sample plan was also guaranteed in order to avoid distortions in the measurements.

Nine samples were prepared, one for each capillary tube, following the recommendations of Pottker and Stähelin, (2001). Four pieces of capillary tube, two of each extremity, were embedded in each sample. The measurements were taken using the tube pieces with the best quality only, with one of each extremity. Six measurements were taken for each capillary tube using a microscopic with laser beams in the following angular positions: 0°, 30°, 60°, 90°, 120° and 150°. The capillary tube inner diameter corresponded to the arithmetic mean of the twelve measurements. This process guaranteed a maximum uncertainty of ±15 μm.

## 5. EXPERIMENTAL PLANNING

The experiments were planned considering four independent variables; two geometric (internal diameter (dc) and length (Lc) of the capillary tube and two operational (degree of subcooling (ΔTsub) and inlet pressure (Pc)). The mass flow rate ( $\dot{m}$ ) was taken as the dependent variable. All of the experiments were performed under choked flow conditions, as a way of eliminating the influence of the evaporating pressure.

To rationalize the number of experiments a statistical procedure, based on a complete and mixed 2 & 3 level factorial design (Box et al., 1978), was employed. Such a technique allows the evaluation of the effect of each independent variable, as well as the effect of the interactions among them on the dependent variable.

The experimental results were correlated using dimensionless parameters. The dimensionless parameters were obtained from the Buckingham-Pi Theorem (Fox and McDonald, 1998), considering the mass flow rate as the dependent parameter and the mass [M], length [L] and time [t] as the primary dimensions The internal diameter of the capillary tube ($d_c$) and the refrigerant viscosity ($\mu_f$) and density ($\rho_f$) were considered as the repeating parameters. The resulting dimensionless parameters were as follows:

$$\pi_1 = \dot{m} \big/ d_c\, \mu_f \qquad (1)$$

$$\pi_2 = d_c^2 \rho_f P_c \big/ \mu_f^2 \qquad (2)$$

$$\pi_3 = L_c \big/ d_c \qquad (3)$$

$$\pi_4 = d_c^2 \rho_f^2 c_p \Delta T_{sub} \big/ \mu_f^2 \qquad (4)$$

The subcooling for the refrigerant mixtures R-507a, R-404a and R-407c was defined as the difference between the bubble point temperature and the fluid temperature at the inlet of the capillary tube. For these fluids the inlet pressure was taken as the saturation pressure corresponding to the bubble point temperature. The refrigerant properties were evaluated by the REFPROP 7.0 program (McLinden et al. 2001), using the fluid temperature at the inlet of the capillary tube as the reference condition.

The dimensionless parameters were correlated in a power law form (Bittle et al., 1998):

$$\pi_1 = f . \pi_2^a . \pi_3^b . \pi_4^c \qquad (5)$$

## 6. EXPERIMENTAL RESULTS

The commercialization of R-22 has been restricted due to ozone depletion and global warming concerns. However as this substance is still widely used in small and medium capacity refrigeration systems, it was tested to serve as a baseline for the other fluids.

In the R-22 tests, both the internal diameter and the tube length were considered as 3 level factorial design variables. The other input data were considered as 2 level factorial design variables. This gave a total of 64 tests, 28 of them being repeated tests (see Table 1).

The repeated tests are required for the effects analysis (Box et al., 1978). Such analysis allows the identification of the parameters or the interactions among them, with the greatest impact on the dependent variable. Empiric correlations based on this technique have also been developed (Melo et al., 2003), but are out of the scope of this work.

Table 1: Experimental results for R-22

| Test | $P_c$ [bar] | $\Delta T_{Sub}$ [°C] | $d_c$ [mm] | $L_c$ [mm] | $\dot{m}$ [kg/h] |
|---|---|---|---|---|---|
| 1 | 15.35 | 5.1 | 1.012 | 2000 | 11.7 |
| 2 | 19.46 | 5.1 | 1.012 | 2000 | 13.6 |
| 3 | 15.37 | 10.1 | 1.012 | 2000 | 13.4 |
| 4 | 19.46 | 10.0 | 1.012 | 2000 | 15.2 |
| 5 | 15.38 | 5.0 | 1.487 | 1999 | 37.3 |
| 6 | 19.46 | 5.0 | 1.487 | 1999 | 42.8 |
| 7 | 15.35 | 10.0 | 1.487 | 1999 | 42.2 |
| 8 | 19.46 | 10.0 | 1.487 | 1999 | 47.8 |
| 9 | 15.37 | 5.0 | 1.487 | 1999 | 36.8 |
| 10 | 19.48 | 5.1 | 1.487 | 1999 | 42.7 |
| 11 | 15.36 | 10.0 | 1.487 | 1999 | 42.1 |
| 12 | 19.47 | 10.0 | 1.487 | 1999 | 47.6 |
| 13 | 15.35 | 5.0 | 2.213 | 2000 | 104.2 |
| 14 | 19.51 | 5.0 | 2.213 | 2000 | 117.4 |
| 15 | 15.37 | 10.0 | 2.213 | 2000 | 117.2 |
| 16 | 19.48 | 10.0 | 2.213 | 2000 | 130.5 |
| 17 | 15.35 | 5.2 | 0.993 | 3000 | 9.4 |
| 18 | 19.47 | 5.0 | 0.993 | 3000 | 11.1 |
| 19 | 15.33 | 9.9 | 0.993 | 3000 | 10.7 |
| 20 | 19.43 | 10.0 | 0.993 | 3000 | 12.3 |
| 21 | 15.40 | 5.0 | 1.501 | 3000 | 30.4 |
| 22 | 19.48 | 5.0 | 1.501 | 3000 | 35.1 |
| 23 | 15.37 | 10.0 | 1.501 | 3000 | 34.4 |
| 24 | 19.47 | 10.1 | 1.501 | 3000 | 38.4 |
| 25 | 15.39 | 5.0 | 1.501 | 3000 | 30.5 |
| 26 | 19.51 | 5.0 | 1.501 | 3000 | 35.3 |
| 27 | 15.37 | 10.0 | 1.501 | 3000 | 33.5 |
| 28 | 19.46 | 10.1 | 1.501 | 3000 | 38.4 |
| 29 | 15.36 | 5.0 | 2.219 | 3000 | 89.9 |
| 30 | 19.45 | 5.0 | 2.219 | 3000 | 101.0 |
| 31 | 15.36 | 10.0 | 2.219 | 3000 | 102.5 |
| 32 | 19.47 | 10.0 | 2.219 | 3000 | 110.5 |
| 33 | 15.36 | 5.0 | 0.993 | 3000 | 9.5 |
| 34 | 19.46 | 5.1 | 0.993 | 3000 | 11.1 |
| 35 | 15.37 | 9.9 | 0.993 | 3000 | 10.7 |
| 36 | 19.47 | 10.1 | 0.993 | 3000 | 12.2 |
| 37 | 15.37 | 5.1 | 1.501 | 3000 | 31.1 |
| 38 | 19.50 | 5.1 | 1.501 | 3000 | 35.0 |
| 39 | 15.38 | 10.0 | 1.501 | 3000 | 34.3 |
| 40 | 19.45 | 10.0 | 1.501 | 3000 | 38.6 |
| 41 | 15.36 | 5.0 | 1.501 | 3000 | 30.4 |
| 42 | 19.47 | 5.0 | 1.501 | 3000 | 35.1 |
| 43 | 15.33 | 10.0 | 1.501 | 3000 | 34.4 |
| 44 | 19.48 | 10.0 | 1.501 | 3000 | 39.1 |
| 45 | 15.36 | 5.0 | 2.219 | 3000 | 89.1 |
| 46 | 19.51 | 5.0 | 2.219 | 3000 | 100.4 |
| 47 | 15.38 | 10.0 | 2.219 | 3000 | 102.1 |
| 48 | 19.47 | 10.0 | 2.219 | 3000 | 109.3 |
| 49 | 15.48 | 5.3 | 1.003 | 4000 | 8.0 |
| 50 | 19.47 | 5.1 | 1.003 | 4000 | 9.3 |
| 51 | 15.36 | 10.2 | 1.003 | 4000 | 9.3 |
| 52 | 19.46 | 9.9 | 1.003 | 4000 | 10.6 |
| 53 | 15.30 | 5.1 | 1.495 | 4000 | 26.5 |
| 54 | 19.48 | 5.0 | 1.495 | 4000 | 31.1 |
| 55 | 15.37 | 10.0 | 1.495 | 4000 | 28.3 |
| 56 | 19.50 | 10.0 | 1.495 | 4000 | 34.5 |
| 57 | 15.41 | 5.3 | 1.495 | 4000 | 26.5 |
| 58 | 19.45 | 5.0 | 1.495 | 4000 | 30.8 |
| 59 | 15.35 | 10.1 | 1.495 | 4000 | 29.0 |
| 60 | 19.49 | 10.0 | 1.495 | 4000 | 34.4 |
| 61 | 15.35 | 5.0 | 2.212 | 3994 | 74.6 |
| 62 | 19.46 | 5.0 | 2.212 | 3994 | 88.5 |
| 63 | 15.35 | 10.0 | 2.212 | 3994 | 87.3 |
| 64 | 19.50 | 10.0 | 2.212 | 3994 | 98.5 |

The refrigerant R-507a is an azeotropic mixture of 50% HFC-125 with 50% HFC-143a. Like R-22, R-507a is not a toxic substance, but requires synthetic lubricants. In all tests with refrigerant mixtures the length of the capillary tube was considered as a 2 level factorial design variable. This reduced the number of tests to 32, 8 of them being repeated, and didn't limit the results generality. The results for R-507a are shown in Table 2.

The refrigerant R-404a is a near azeotropic mixture of the refrigerants R-125 (44%), R-143a (52%) and R-134a (4%). The composition of the liquid and vapor phases of this mixture are different, requiring a special care in relation to the charging process of the refrigeration loop and also with leakages. The greatest temperature variation of this fluid during the phase change process is about 1°C, in the pressure range from 0.8 to 26.0 bar. The experimental results obtained with R-404a are shown in Table 3.

Table 2: Experimental results for R-507a

| Test | $P_c$ [bar] | $\Delta T_{Sub}$ [°C] | $d_c$ [mm] | $L_c$ [mm] | $\dot{m}$ [kg/h] |
|---|---|---|---|---|---|
| 1 | 18.76 | 5.1 | 1.012 | 2000 | 13.6 |
| 2 | 23.63 | 5.0 | 1.012 | 2000 | 15.0 |
| 3 | 18.77 | 10.0 | 1.012 | 2000 | 14.1 |
| 4 | 23.67 | 10.1 | 1.012 | 2000 | 16.9 |
| 5 | 18.81 | 5.1 | 1.003 | 4000 | 9.3 |
| 6 | 23.69 | 5.0 | 1.003 | 4000 | 10.8 |
| 7 | 18.70 | 10.0 | 1.003 | 4000 | 10.1 |
| 8 | 23.67 | 10.0 | 1.003 | 4000 | 11.8 |
| 9 | 18.75 | 5.1 | 1.487 | 1999 | 40.7 |
| 10 | 23.65 | 5.0 | 1.487 | 1999 | 45.8 |
| 11 | 18.75 | 10.0 | 1.487 | 1999 | 44.6 |
| 12 | 23.65 | 10.0 | 1.487 | 1999 | 49.7 |
| 13 | 18.75 | 5.0 | 1.495 | 4000 | 29.1 |
| 14 | 23.69 | 5.0 | 1.495 | 4000 | 33.3 |
| 15 | 18.80 | 10.1 | 1.495 | 4000 | 31.3 |
| 16 | 23.68 | 10.0 | 1.495 | 4000 | 35.5 |

| Test | $P_c$ [bar] | $\Delta T_{Sub}$ [°C] | $d_c$ [mm] | $L_c$ [mm] | $\dot{m}$ [kg/h] |
|---|---|---|---|---|---|
| 17 | 18.75 | 5.1 | 1.487 | 1999 | 39.2 |
| 18 | 23.68 | 5.0 | 1.487 | 1999 | 44.4 |
| 19 | 18.74 | 10.0 | 1.487 | 1999 | 42.7 |
| 20 | 23.65 | 10.0 | 1.487 | 1999 | 50.2 |
| 21 | 18.77 | 5.1 | 1.495 | 4000 | 30.0 |
| 22 | 23.66 | 5.1 | 1.495 | 4000 | 32.9 |
| 23 | 18.76 | 10.0 | 1.495 | 4000 | 32.6 |
| 24 | 23.63 | 10.0 | 1.495 | 4000 | 36.2 |
| 25 | 18.70 | 3.6 | 2.213 | 2000 | 110.3 |
| 26 | 23.65 | 5.0 | 2.213 | 2000 | 124.9 |
| 27 | 18.75 | 10.0 | 2.213 | 2000 | 124.2 |
| 28 | 23.66 | 10.1 | 2.213 | 2000 | 140.0 |
| 29 | 18.73 | 5.0 | 2.212 | 3994 | 83.5 |
| 30 | 23.66 | 5.0 | 2.212 | 3994 | 94.1 |
| 31 | 18.77 | 9.9 | 2.212 | 3994 | 92.6 |
| 32 | 23.67 | 10.0 | 2.212 | 3994 | 103.4 |

Table 3: Experimental results for R-404a

| Test | $P_c$ [bar] | $\Delta T_{Sub}$ [°C] | $d_c$ [mm] | $L_c$ [mm] | $\dot{m}$ [kg/h] |
|---|---|---|---|---|---|
| 1 | 18.29 | 5.1 | 1.012 | 2000 | 11.6 |
| 2 | 23.08 | 4.9 | 1.012 | 2000 | 13.6 |
| 3 | 18.29 | 10.1 | 1.012 | 2000 | 12.4 |
| 4 | 23.04 | 10.0 | 1.012 | 2000 | 14.4 |
| 5 | 18.31 | 5.0 | 1.003 | 4000 | 7.9 |
| 6 | 23.04 | 4.9 | 1.003 | 4000 | 9.6 |
| 7 | 18.29 | 9.9 | 1.003 | 4000 | 8.6 |
| 8 | 23.03 | 9.8 | 1.003 | 4000 | 10.3 |
| 9 | 18.28 | 5.0 | 1.487 | 1999 | 40.2 |
| 10 | 23.08 | 5.0 | 1.487 | 1999 | 42.2 |
| 11 | 18.29 | 10.0 | 1.487 | 1999 | 37.7 |
| 12 | 23.07 | 9.7 | 1.487 | 1999 | 46.5 |
| 13 | 18.26 | 5.0 | 1.495 | 4000 | 27.5 |
| 14 | 23.05 | 4.8 | 1.495 | 4000 | 30.8 |
| 15 | 18.28 | 10.0 | 1.495 | 4000 | 29.9 |
| 16 | 23.07 | 9.9 | 1.495 | 4000 | 33.9 |

| Test | $P_c$ [bar] | $\Delta T_{Sub}$ [°C] | $d_c$ [mm] | $L_c$ [mm] | $\dot{m}$ [kg/h] |
|---|---|---|---|---|---|
| 17 | 18.29 | 5.0 | 1.487 | 1999 | 37.0 |
| 18 | 23.09 | 4.9 | 1.487 | 1999 | 42.9 |
| 19 | 18.29 | 10.0 | 1.487 | 1999 | 40.4 |
| 20 | 23.08 | 9.9 | 1.487 | 1999 | 46.7 |
| 21 | 18.28 | 5.2 | 1.495 | 4000 | 27.2 |
| 22 | 23.06 | 5.0 | 1.495 | 4000 | 30.6 |
| 23 | 18.28 | 10.0 | 1.495 | 4000 | 29.6 |
| 24 | 23.05 | 10.0 | 1.495 | 4000 | 33.1 |
| 25 | 18.29 | 5.1 | 2.213 | 2000 | 108.2 |
| 26 | 22.99 | 5.0 | 2.213 | 2000 | 121.5 |
| 27 | 18.26 | 10.0 | 2.213 | 2000 | 124.1 |
| 28 | 23.04 | 9.9 | 2.213 | 2000 | 136.4 |
| 29 | 18.28 | 5.0 | 2.212 | 3994 | 81.4 |
| 30 | 23.04 | 4.9 | 2.212 | 3994 | 90.8 |
| 31 | 18.29 | 10.0 | 2.212 | 3994 | 92.1 |
| 32 | 23.06 | 10.0 | 2.212 | 3994 | 101.2 |

The refrigerant R-407c is a zeotropic mixture of the refrigerants R-32 (23%), R-125 (25%) and R-134a (52%). The greatest temperature variation of this fluid during the phase change process was around 7°C, in the pressure range from 0.5 to 23.0 bar, it being always higher than 4°C. The experimental results obtained with R-407c are shown in Table 4.

The experimental results were used to develop a dimensionless correlation as indicated by equations (1) to (5). The equation coefficients for each of the refrigerants are given in Table 5. The maximum relative deviation between measured and correlated data was 5.8%, 3.2%, 7.4% and of 6.2%, respectively for the refrigerants R-22, R-507a, R-404a, and R-407c.

Table 5: Equation (5) coefficients

| Fluid | a | b | c | f |
|---|---|---|---|---|
| R-22 | 0.483984 | -0.482766 | 0.168552 | 0.051147 |
| R-507a | 0.504488 | -0.468537 | 0.134711 | 0.067283 |
| R-404a | 0.553862 | -0.473304 | 0.147468 | 0.012243 |
| R-407c | 0.502478 | -0.467535 | 0.183570 | 0.017421 |

Table 4: Experimental results for R-407c

| Test | $P_c$ [bar] | $\Delta T_{Sub}$ [°C] | $d_c$ [mm] | $L_c$ [mm] | $\dot{m}$ [kg/h] |
|---|---|---|---|---|---|
| 1 | 17.38 | 4.8 | 1.012 | 2000 | 11.7 |
| 2 | 22.23 | 5.2 | 1.012 | 2000 | 13.6 |
| 3 | 17.40 | 9.9 | 1.012 | 2000 | 13.2 |
| 4 | 22.17 | 10.1 | 1.012 | 2000 | 14.8 |
| 5 | 17.37 | 4.8 | 1.003 | 4000 | 8.4 |
| 6 | 22.19 | 5.3 | 1.003 | 4000 | 9.6 |
| 7 | 17.34 | 9.6 | 1.003 | 4000 | 9.5 |
| 8 | 22.25 | 10.2 | 1.003 | 4000 | 10.7 |
| 9 | 17.41 | 4.9 | 1.487 | 1999 | 39.1 |
| 10 | 22.23 | 5.2 | 1.487 | 1999 | 43.9 |
| 11 | 17.39 | 9.7 | 1.487 | 1999 | 43.0 |
| 12 | 22.24 | 10.2 | 1.487 | 1999 | 48.3 |
| 13 | 17.40 | 4.9 | 1.495 | 4000 | 27.8 |
| 14 | 22.20 | 5.2 | 1.495 | 4000 | 32.4 |
| 15 | 17.40 | 9.9 | 1.495 | 4000 | 30.9 |
| 16 | 22.20 | 10.2 | 1.495 | 4000 | 35.1 |
| 17 | 17.40 | 5.0 | 1.487 | 1999 | 39.1 |
| 18 | 22.19 | 5.2 | 1.487 | 1999 | 44.2 |
| 19 | 17.36 | 9.8 | 1.487 | 1999 | 43.3 |
| 20 | 22.23 | 10.2 | 1.487 | 1999 | 48.4 |
| 21 | 17.38 | 4.9 | 1.495 | 4000 | 27.6 |
| 22 | 22.22 | 5.4 | 1.495 | 4000 | 32.4 |
| 23 | 17.40 | 9.9 | 1.495 | 4000 | 30.5 |
| 24 | 22.20 | 10.1 | 1.495 | 4000 | 34.9 |
| 25 | 17.40 | 4.8 | 2.213 | 2000 | 107.1 |
| 26 | 22.19 | 5.2 | 2.213 | 2000 | 124.0 |
| 27 | 17.43 | 10.0 | 2.213 | 2000 | 123.3 |
| 28 | 22.23 | 10.1 | 2.213 | 2000 | 138.3 |
| 29 | 17.39 | 4.8 | 2.212 | 3994 | 81.4 |
| 30 | 22.23 | 5.3 | 2.212 | 3994 | 93.9 |
| 31 | 17.39 | 10.0 | 2.212 | 3994 | 95.2 |
| 32 | 22.21 | 10.2 | 2.212 | 3994 | 102.6 |

## 6. COMPARISON WITH THE ASHRAE CORRELATION

ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers) (ASHRAE, 2002), based on experimental data obtained with the refrigerants R-134a, R-22 and R-410a, has published a generalized correlation to predict the refrigerant flow through capillary tubes, using eight dimensionless parameters. This equation was fed with the measured data from this study and the comparisons are exhibited in Figure 6.

As can be seen the relative deviation between the measured and ASHRAE's predicted data is highly sensitive to the mass flow rate. With the exception of R-407c, the ASHRAE design equation underestimates the measured data by up to 20%, for mass flow rates higher than 80 kg/h. For mass flow rates lower than 20kg/h, the equation overestimates the measured data, especially for the refrigerants R-404a and R-407c, where deviations of up to 25% are observed.

## 7. CONCLUDING REMARKS

The development and application of alternative refrigerants due to environmental concerns regarding the depletion of the ozone layer and global warming are current technological needs. Procedures for estimating the geometric characteristics of a capillary tube for a given application are also required. This work approached these two aspects, providing measured data for refrigerant fluids with little ecological impact and also providing correlations that expressed the relationship among the geometric and operational data, the fluid type and the refrigerant mass flow rate.

The experiments addressed the commercial refrigeration area, including mass flow rates ranging from 8 to 140 kg/h. The maximum relative deviations between the predicted and measured mass flow rates were from 3.2% to 7.4%, depending on the refrigerant type.

The ASHRAE correlation provided relative deviations considerably higher than the those obtained in this study. Furthermore the prediction capability of ASHRAE's equation was strongly dependent on the mass flow rate. The introduction of additional dimensionless terms, to take fluid properties into account, and thereby generate a generalized correlation, seems to be an inappropriate strategy.

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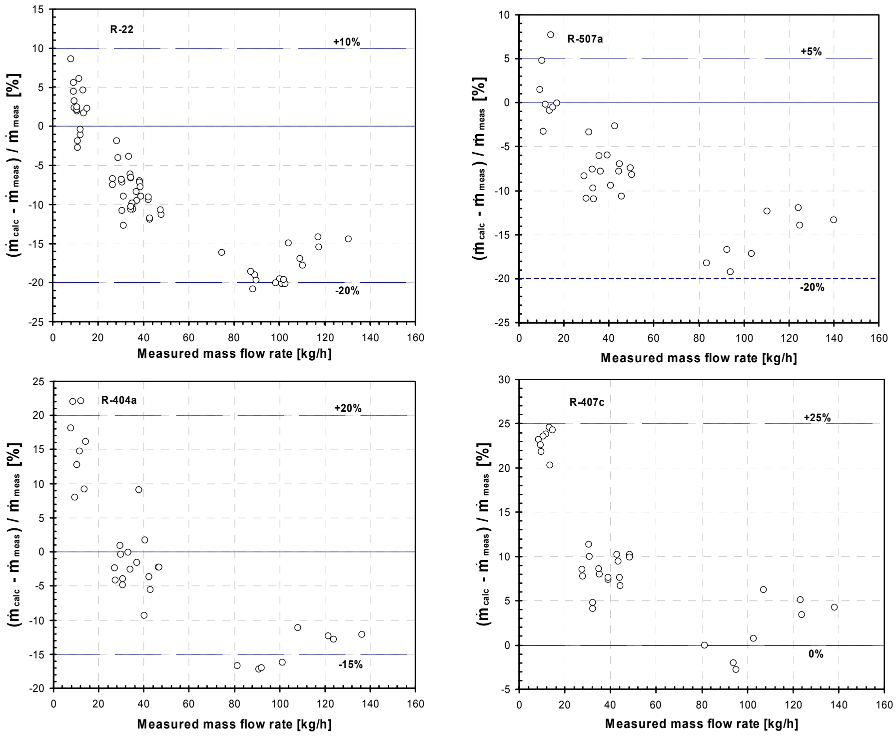


Figure 6: Comparisons with ASHRAE's correlation

## ACKNOWLEDGEMENTS

The authors would like to thank Embraco S.A. for sponsoring this research work, in particular Messrs. M. R. Thiessen and G. B. Melo. The financial support from the Brazilian funding agencies *CNPq* and *CAPES* are also duly acknowledged.