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EXPERIMENTAL STUDIES ON NON-ADIABATIC FLOW OF HFC-134a THROUGH CAPILLARY TUBES

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ABSTRACT

This paper discusses results from an experimental work on concentric capillary tube-suction line heat exchangers commonly used as refrigerant control devices in household refrigerators and freezers. Heat exchanger performance with the hydrofluorcarbon R-134a was experimentally evaluated for a range of heat exchanger geometries (diameter and length of the capillary tube, length and position of the heat exchanger) and operating boundary conditions (condensing pressure and subcooling). The results provided enough data to study the influence of the independent variables (geometry and operating conditions) on the dependent variables (refrigerant mass flow rate and temperature profiles).

INTRODUCTION

Nowadays every household refrigerating systems employs a capillary tube for metering the flow of liquid refrigerant. It is simply a small bore tube connecting the condenser to the evaporator. Liquid refrigerant flows into one end and expands down to the evaporator pressure. In doing so it meters refrigerant at the desired mass flow rate. It usually forms a counterflow heat exchanger with the suction line, in order to increase evaporator capacity and to prevent slugging of the compressor and sweating of the suction line. The refrigerant flow through the capillary tube may be divided into three regions: a heat exchange (diabatic) region and two adiabatic regions, one upstream and other downstream of the heat exchanger (see Figure 1).

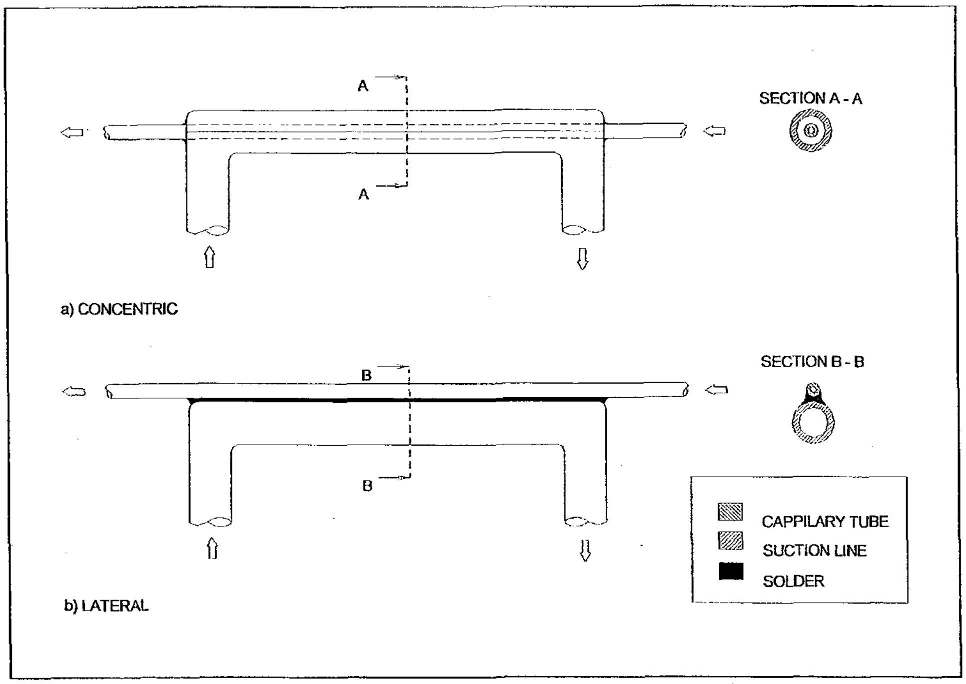

Figure 1 – Concentric and lateral heat exchangers

Two types of heat exchanger are usually found: lateral and concentric. Lateral heat exchangers are the ones where the capillary tube is soldered to the suction line whereas concentric heat exchangers are the ones where the capillary tube passes inside the suction line. The vast majority of previous non-adiabatic capillary tube performance work was carried out with lateral capillary tube-suction line heat exchangers (Pate and Tree, 1984) (Bittle et al., 1995) in despite of the widespread use of concentric heat exchangers in many parts of the world.

The study presented herein focuses on an experimental evaluation of concentric capillary tube-suction line heat exchanger performance with HFC-134a. Several capillary tube length and diameter combinations were tested with different heat exchanger lengths and positions using a test loop that provided full control of the boundary operating conditions (Zangari, 1998). The analysis of the collected data base provided significant insight into the behavior of a concentric capillary tube-suction line heat exchanger and also provided enough data to explore the relationship between the independent (geometry and operating conditions) and dependent (mass flow rate and temperature profiles) variables.

TEST SECTION

The experimental apparatus developed by Gonçalves, 1994, and Mendonça, 1996, was used to determine the flow characteristics of various capillary tube-suction line heat exchangers. A new test section was developed and is schematically shown in Figure 2.

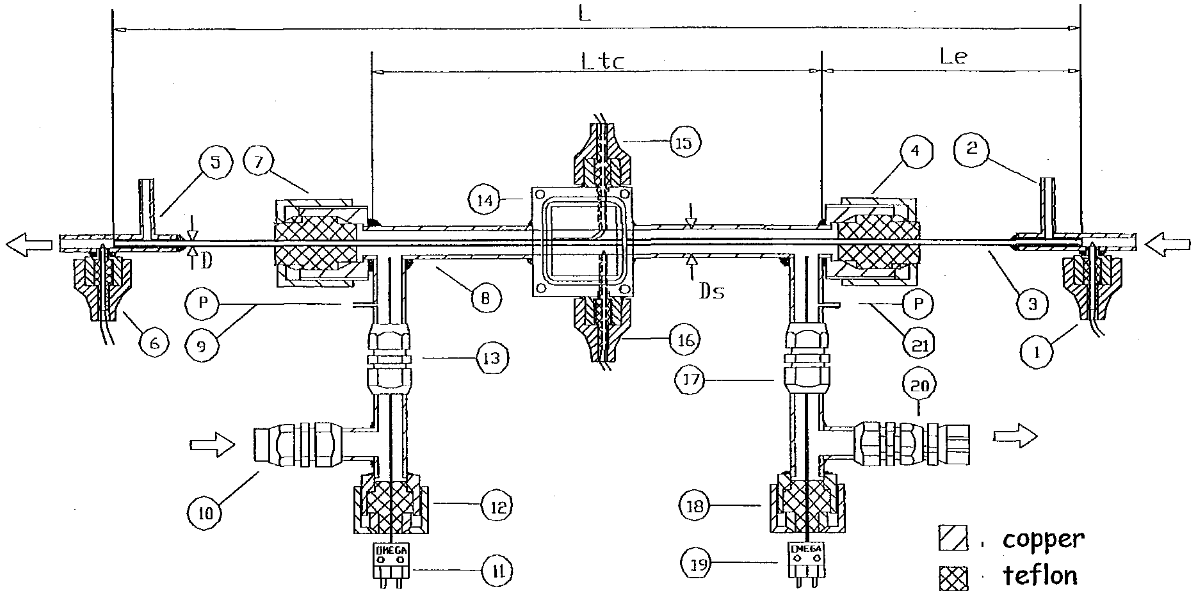

Figure 2 – Test section

The geometrical characteristics of the capillary tube-suction line heat exchanger (length, L, and diameter, D, of the capillary tube, heat exchanger length, Ltc, adiabatic inlet length, Le, diameter of the suction line, Ds) are indicated in Figure 2. The measurement stations for the boundary operating conditions (inlet (2) and evaporating (9) pressures, refrigerant temperature at the inlet (11) and outlet (19) of the suction line (11)) are also indicated in Figure 2. The subcooling was calculated from the temperature (1) and pressure (2) measurements at the inlet of the capillary tube. Additional measurements stations for the refrigerant temperature (6) and pressure (5) at the outlet of the capillary tube and for the refrigerant pressure at the outlet of the suction line (21) are also shown in Figure 2. The refrigerant temperature at the inlet and outlet of the suction line was measured by T-type immersion thermocouples in order to avoid heat conduction errors as pointed out by Mendonça, 1996. Thermocouples were also placed along the length of the capillary tube and suction line to determine the wall temperature distribution. A special measuring station (14) for the suction line fluid temperature (16) and capillary tube wall temperature (15) in the diabatic region is also shown in Figure 2. Further details of the instrumentation are given in Zangari, 1998.

TYPICAL TEMPERATURE PROFILES

Figure 3 shows a typical temperature distribution for a concentric capillary tube-suction line heat exchanger. The symbols Pe, ΔT, Te, Tout, Tin, and $\dot{m}$ denote inlet pressure, subcooling, evaporating temperature, suction line outlet temperature, suction line inlet temperature and refrigerant mass flow rate, respectively. No distinction was made between the capillary tube wall and fluid temperatures due to the high convective heat transfer coefficient on the capillary side. Both the suction line fluid and wall temperatures were measured.

D = 0.61 mm, L = 3.0 m, Ds = 7.86 mm, Ltc = 2.20 m, Le = 0.20m
Pe = 13.97 bar; ΔT = 10.2 °C; Te = -23,0°C; Tin = -11.0 °C; Tout = 26.0°C ; $\dot{m}$ =3.26 kg/h

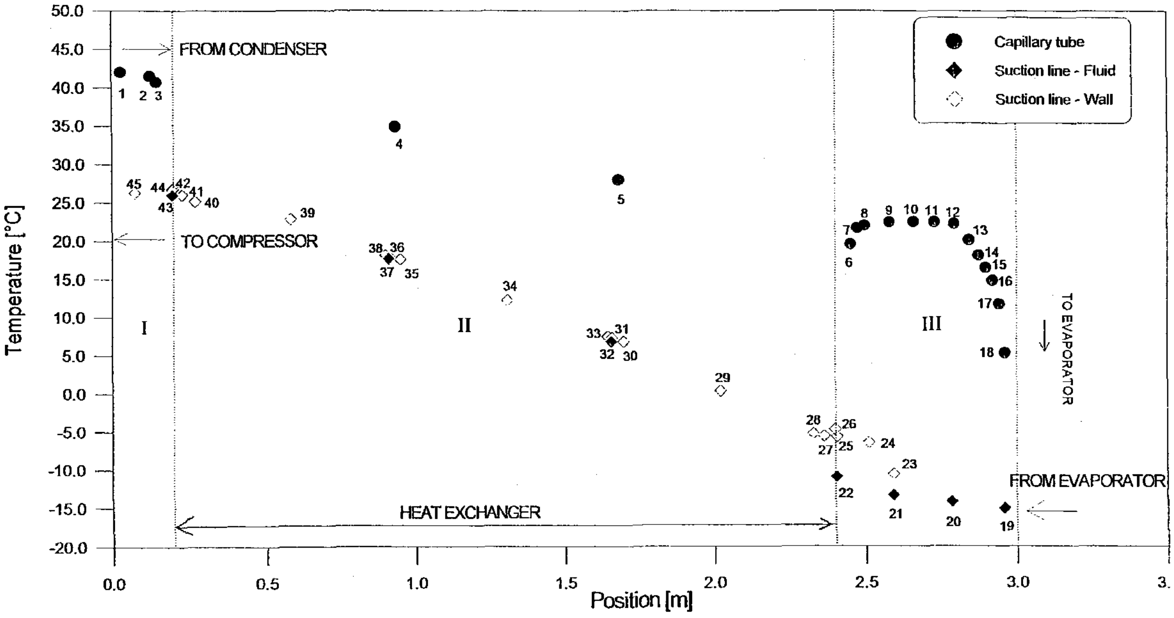

Figure 3 – Typical temperature profiles along the capillary tube and suction line.

The capillary tube wall temperature remains almost constant along the adiabatic inlet region (I). The observed dip in wall temperature between points 2 and 3 results from the heat transfer from the capillary (warm) to the suction line (cold) walls. The contact between the suction line and the capillary tube in both ends of the heat exchanger was avoided, but the teflon plug was not thick enough to avoid the heat transfer to the suction line. The refrigerant in the capillary remains as liquid along the heat exchanger region. The observed temperature drop in this region is the result of a heat transfer to the suction line. At the limit between the diabatic (II) and adiabatic outlet (III) regions the temperature difference between the capillary and suction line walls reaches a maximum value. This leads to a considerable conduction heat transfer to the suction line wall, decreasing the capillary wall temperature (point 6). The influence of the suction line acting as a heat sink disappears at point (8). The very deep temperature drop at point (12) is a clear indication of the beginning of the vaporization process.

Fluid temperature measurements were also taken along the inlet of the suction line (points 19-21). It can be observed that there is a small temperature gradient between points 19 and 22 and also between points 21 and 22. The temperature gradient between points 19 and 20 is mostly due to the heat gain from the surrounding air whereas the temperature gradient between points 21 and 22 is partly due to the heat gain from the ambient air and partly due to the conduction heat transfer between the capillary tube and suction line. The already mentioned temperature difference between points 26 and 22 is then also partly due to the heat exchanges with the surrounding air. The suction line fluid temperature at points 31 and 36 were measured using specially manufactured measuring stations (see Figure 2). It can be seen that there is almost no difference between the fluid and wall suction line temperature in the diabatic region. The suction line wall and fluid temperatures are also quite close at the outlet of the suction line.

These results are in line with the ones reported by Mendonça, 1998 for lateral capillary tube-suction line heat exchangers.

EFFECTS OF THE BOUNDARY OPERATING CONDITIONS

Capillary tube and suction line temperature profiles have been plotted in figure 4 for two different inlet pressures, 9 and 14 bar. It can be observed that the shapes of the temperature profiles are practically unaffected by the inlet pressure. As expected the capillary tube temperature profile shifts accordingly to the inlet pressure variation. In both cases the refrigerant remains as liquid along most of the length of the capillary tube, with the flash point moving towards the end of the tube when the pressure changes from 9 to 14 bar. The inlet pressure also affects the mass flow rate and the suction line outlet temperature. For an inlet pressure variation of 5 bar (9 to 14 bar) the suction line outlet temperature increases 5.6°C (9.6 to 15.2 °C) and the mass flow rate increases 1.93 kg/h (5.49 to 7.42 kg/h).

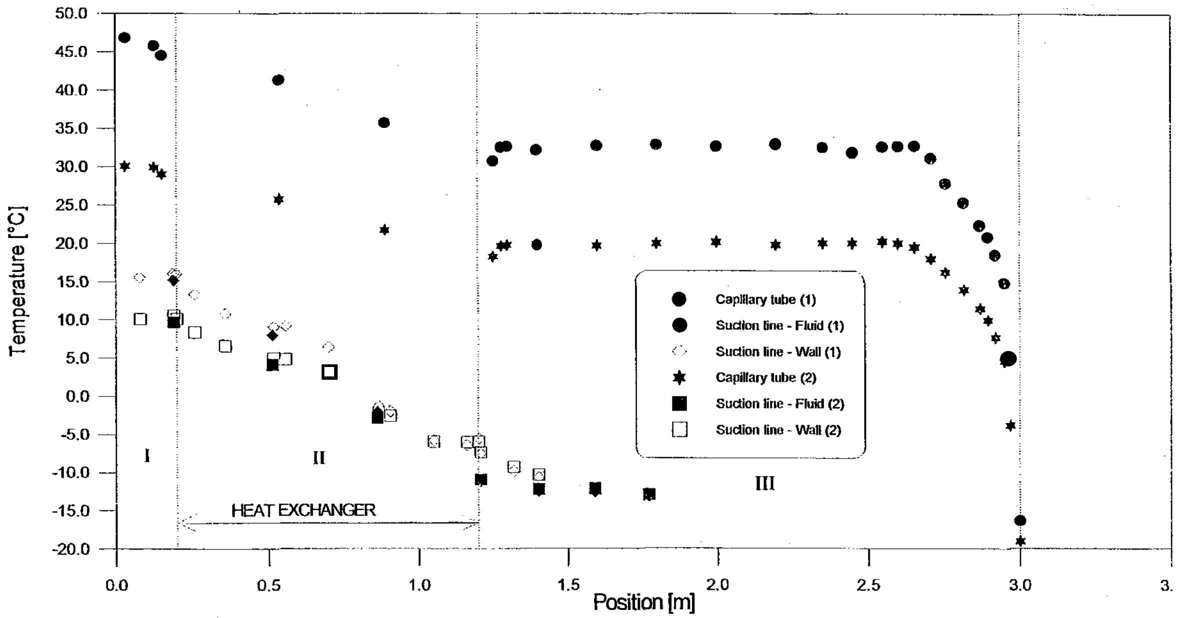
D = 0.83 mm, L = 3.0 m, Ds = 7.86 mm, Ltc = 1.0 m, Le = 0.20m
(1) **Pe = 14.0 bar**; ΔT = 5.0°C; Te = -22.9°C; Tin = -11.2°C; Tout = 15.2°C; $\dot{m}$ = 7.42 kg/h
(2) **Pe = 9.0 bar**; ΔT = 5.2°C; Te = -23.1°C; Tin = -11.0°C; Tout = 9.6°C; $\dot{m}$ = 5.49 kg/h

Figure 4 – Effect of the capillary tube inlet pressure

Plots of the heat exchanger temperature profiles are shown in Figure 5 for two subcooling levels at the inlet of the capillary tube. It can be seen that the shape of the temperature profiles and the flash point are both practically unaffected by subcooling. For a subcooling variation of 5.1°C (5.2 to 10.3 °C) the suction line outlet temperature decreases 2.4°C and the mass flow rate increases 0.34 kg/h.

EFFECTS OF THE GEOMETRY

Plots of the capillary tube and suction line temperatures are shown in Figure 6, for two different diameters capillary tubes. As can be observed the shape of the temperature profiles and the flashing point are almost unaffected by the inner diameter variation. To the contrary, the mass flow rate increases from 3.26 kg/h to 7.92 kg/h with an inner diameter variation of 0.22 mm (0.61 to 0.83 mm).

D = 0.83 mm, L = 3.0 m, Ds = 7.86 mm, Ltc = 1.0 m, Le = 0.20m
(1) Pe = 9.0 bar; ΔT = **10.3°C**; Te = -23.1°C; Tin = -11.2°C; Tout = 7.3°C; $\dot{m}$ = 5.83 kg/h
(2) Pe = 9.0 bar; ΔT = **5.2°C**; Te = -23.1°C; Tin = -11.0°C; Tout = 9.7°C; $\dot{m}$ = 5.49 kg/h

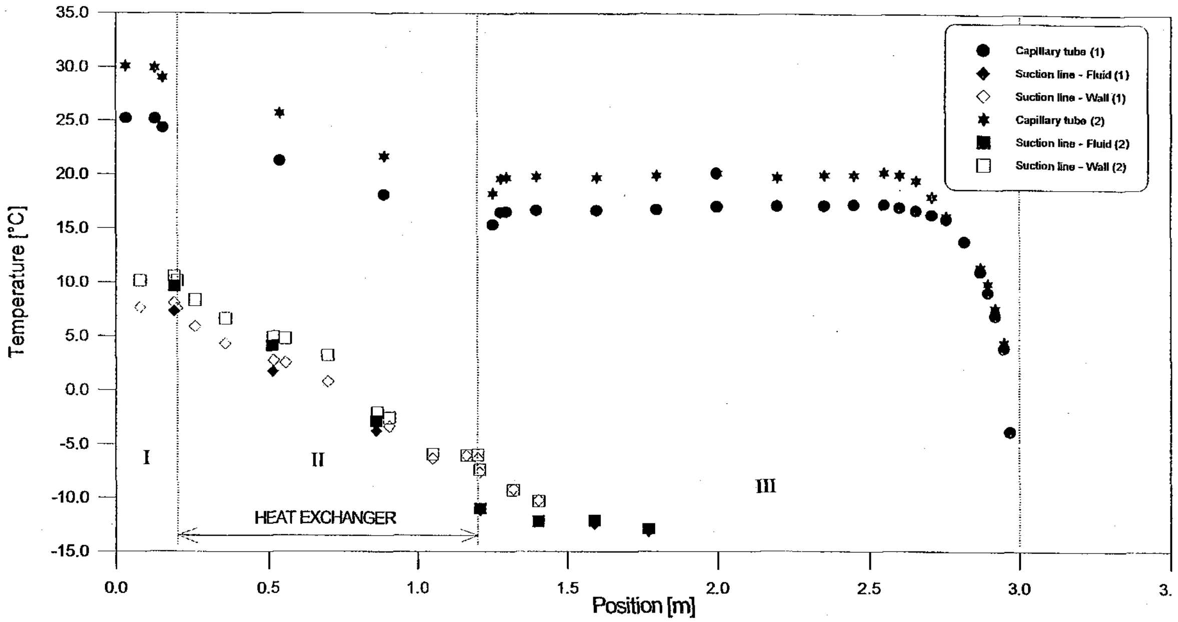

Figure 5 – Effect of subcooling

The effect of the capillary tube length is illustrated in Figure 7. For both lengths the flash point occurs quite close to the end of the capillary tube. For a length variation of 1.0 m (4 to 3 m) the mass flow rate increases 0.35 kg/h (1.91 to 2.26 kg/h).

Plots of the heat exchanger temperature profiles are shown in Figure 8 for two heat exchanger lengths. It can be observed that the mass flow rate is almost unaffected by the heat exchanger length. To the contrary, the flash point is deeply affected by this parameter, moving from almost the end of the capillary tube to the interior of the diabatic region with a length variation of 1.2 m (2.2 to 1.0 m). For a heat exchanger length variation of 1.2 m (1 to 2.2 m) the suction line outlet temperature increases 7.8°C (23.1 to 30.9°C).

Figure 9 explores the positioning effect of the heat exchanger. As can be seen the capillary tube flow pattern is strongly affected by the adiabatic inlet length, changing from a situation where almost only liquid flows through the capillary to a situation where the flash point occurs within the diabatic region. To the contrary, the mass flow rate has an insignificant variation (3.09 to 3.17 kg/h) with an adiabatic inlet length variation of 0.4 m (0.60 to 0.20 m).

(1) **D = 0.61 mm**, L = 3.0 m, Ds = 7.86 mm, Ltc = 2.20 m, Le = 0.20m
Pe = 14.0 bar; ΔT = 10.2°C; Te = -23.0°C; Tin = -10.9°C; Tout = 26.0°C; $\dot{m}$ = 3.26 kg/h
(2) **(D = 0.83 mm)**, L = 3.0 m, Ds = 7.86 mm, Ltc = 2.20 m, Le = 2.20m
Pe = 14.0 bar; ΔT = 10.1°C; Te = -20.8°C; Tin = -11.0°C; Tin = 24.3°C; $\dot{m}$ = 7.92 kg/h

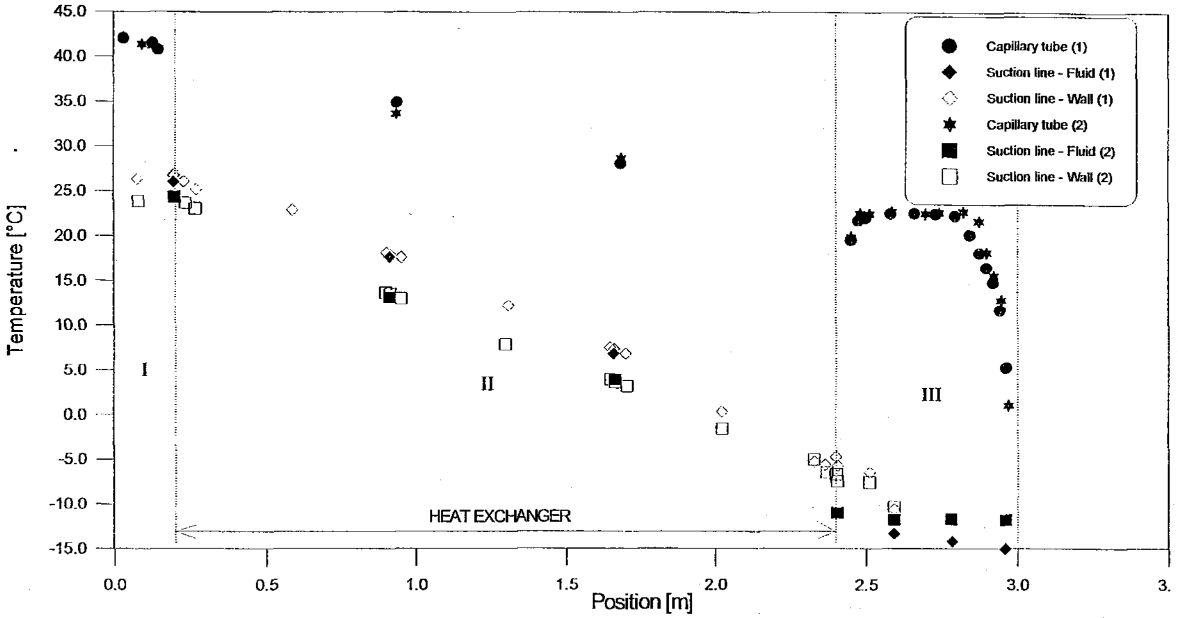

Figure 6 – Effect of the inner diameter of the capillary tube

(1) D = 0.61 mm, **L = 4.0 m**, Ds = 7.86 mm, Ltc = 2.20 m, Le = 0.20m
Pe = 9.0 bar; ΔT = 5.6°C; Te = -23.1°C; Tin = -10.9°C; Tout = 23.4°C; $\dot{m}$ = 1.91 kg/h
(2) D = 0.61 mm, **L = 3.0 m**, Ds = 7.86 mm, Ltc = 2.20 m, Le = 0.20m
Pe = 9.0 bar; ΔT = 5.7°C; Te = -22.9°C; Tin = -11.0°C; Tout = 20.7°C; $\dot{m}$ = 2.26 kg/h

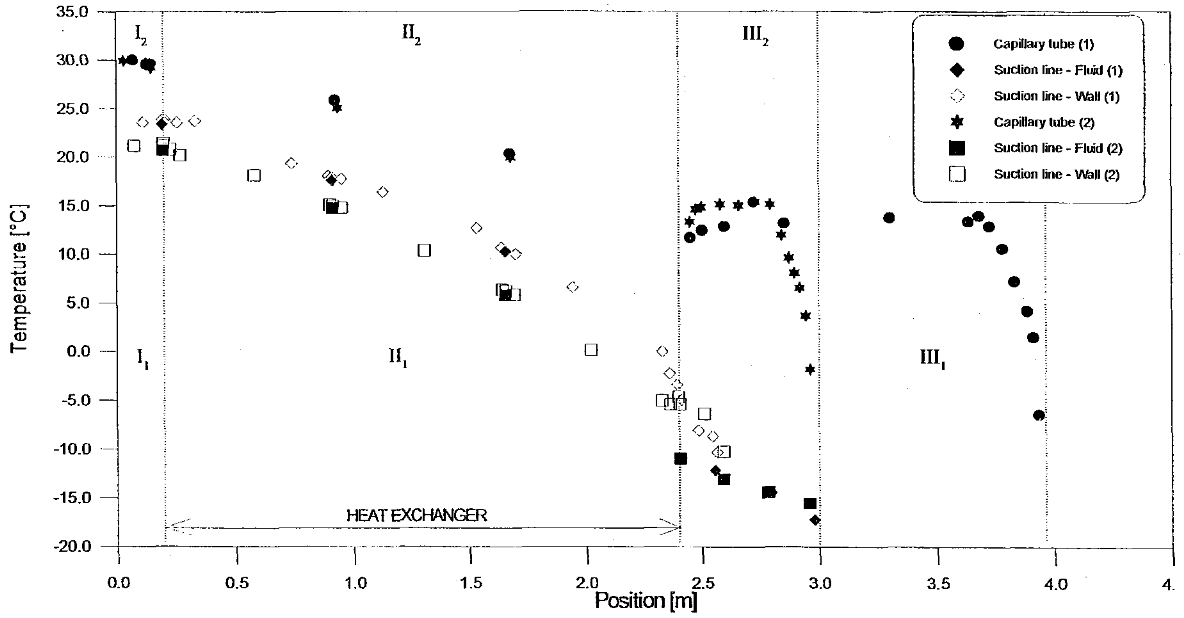

Figure 7 – Effect of the capillary tube length

(1) D = 0.61 mm, L = 3.0 m, Ds = 7.86 mm, **Ltc = 1.0 m**, Le = 0.60m
Pe = 14.0 bar; ΔT = 7.9°C; Te = -23.2°C; Tin = -3.2°C; Tin = 23.1°C; $\dot{m}$ = 2.93 kg/h
(2) D = 0.61 mm, L = 3.0 m, Ds = 7.86 mm, **Ltc = 2.20 m**, Le = 0.60m
Pe = 14.0 bar; ΔT = 8.2°C; Te = -23.0°C; Tin = -29°C; Tout = 30.9°C; $\dot{m}$ = 3.06 kg/h

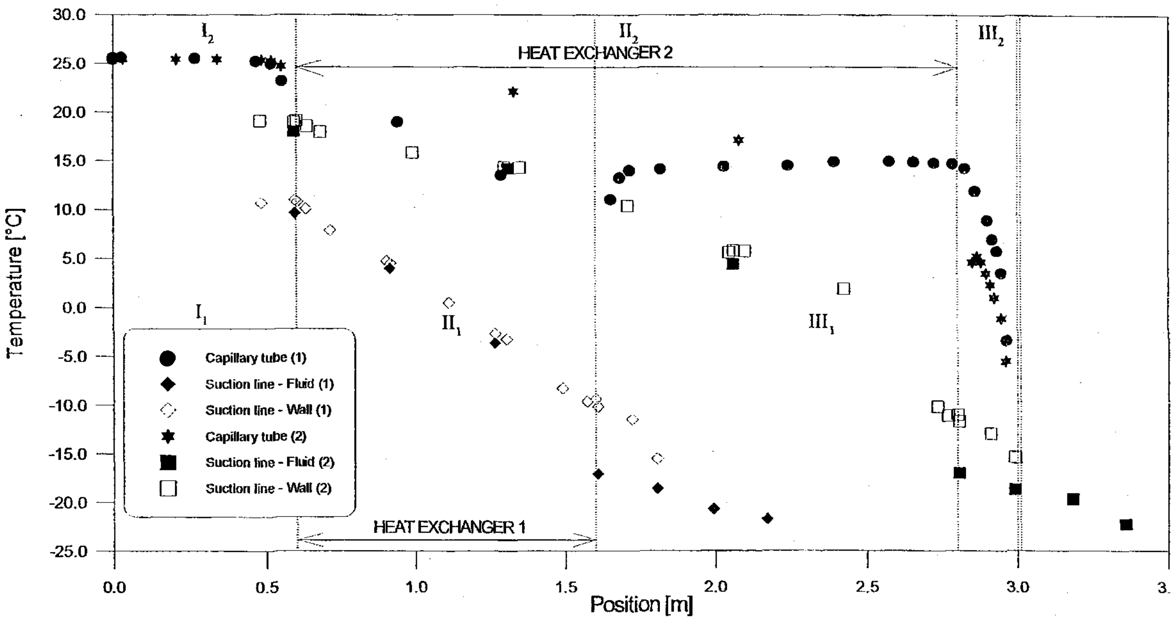

Figure 8 – Effect of the heat exchanger length

(1) D = 0.61 mm, L = 3.0 m, Ds = 7.86 mm, Ltc = 2.20 m, **Le = 0.60m**
Pe = 14.0 bar; ΔT = 5.4°C; Te = -23.0°C; Tin = -17.4°C; Tout = 29.3°C; $\dot{m}$ = 3.09 kg/h
(2) D = 0.61 mm, L = 3.0 m, D_s = 7.86 mm, Ltc = 2.20 m, **Le = 0.20m**
Pe = 14.0 bar; ΔT = 5.7°C; Te = -23.1°C; Tin = -16.9°C; Tout = 27.4°C; $\dot{m}$ = 3.17 kg/h

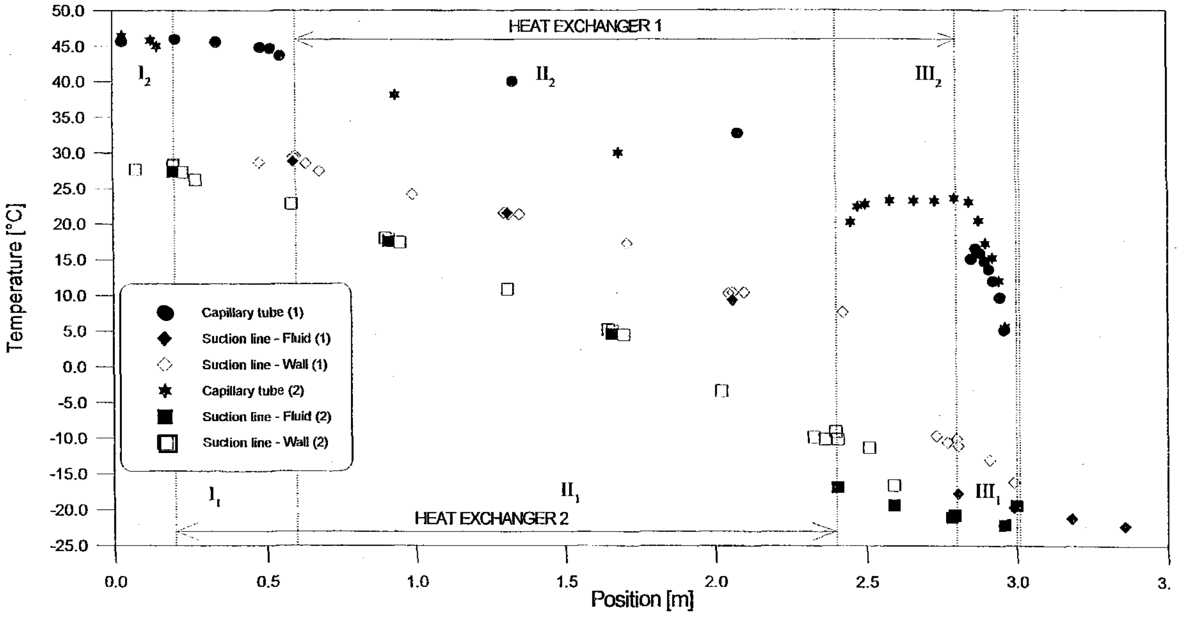

Figure 9 – Effect of the adiabatic inlet length

CONCLUDING REMARKS

A very comprehensive experimental research program dealing with concentric capillary tube-heat exchangers has been performed. Heat exchanger performance with HFC-134a was experimentally evaluated over a wide range of boundary operating conditions and heat exchanger geometries typical of household refrigerators and freezers.

A comparative analysis involving both the operating conditions and the geometrical characteristics of the heat exchangers has also been presented. The operating condition having the greatest effect on the mass flow rate was the inlet pressure. The inner diameter of the capillary tube was the geometrical variable with the greatest impact on the mass flow rate. This parameter has an insignificant effect on the capillary tube and suction line temperature distributions. The effect of the heat exchanger and adiabatic inlet lengths were shown to have a minor effect on the mass flow rate, but did have a significant effect on the suction line outlet temperature.

The resulting database may also be used for comparative studies between adiabatic and non-adiabatic capillary tubes and between lateral and concentric heat exchangers. Validation studies of numerical codes may also be performed.

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