A Design Model for Capillary Tube-Suction Line Heat Exchangers

R. A. Peixoto and C. W. Bullard

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For additional information:

Air Conditioning and Refrigeration Center
University of Illinois
Mechanical & Industrial Engineering Dept.
1206 West Green Street
Urbana, IL 61801

(217) 333-3115

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C. W. Bullard, Principal Investigator

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Introduction

Because of their simplicity and low cost, capillary tubes are used as the expansion device in most small refrigeration and air conditioning systems. Their lack of controllability is partially offset by the fact that charge remains relatively constant in hermetically-sealed systems, as does the temperature lift in many applications. Another advantage is that capillary tubes allow high and low side pressures to equalize during the off-cycle, thereby reducing the starting torque required by the compressor. However the resulting charge migration during the off-cycle can contribute to cycling losses.

For some refrigerants including CFC-12 and HFC-134a, system capacity can be increased by using the cold suction line to lower the enthalpy of the fluid entering the evaporator, with only a modest increase in compressor power. Using a simplified theoretical analysis Domanski et al (1992) demonstrated that suction line-liquid line heat exchange could improve COP for these two refrigerants, but not for HCFC-22. For this reason capillary tube-suction line heat exchangers are used in household refrigerators, while adiabatic capillary tubes are used in room air conditioners.

Figures 1 and 2 illustrate the kinds counterflow capillary tube-suction line heat exchangers (ct-sl hx) used in almost all household refrigerators. It may be formed either soldering the capillary tube on the outside of the suction line or placing the capillary tube inside the suction line.

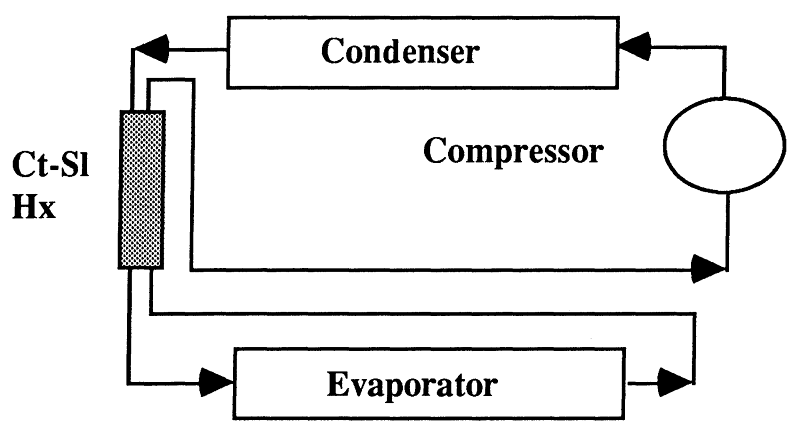
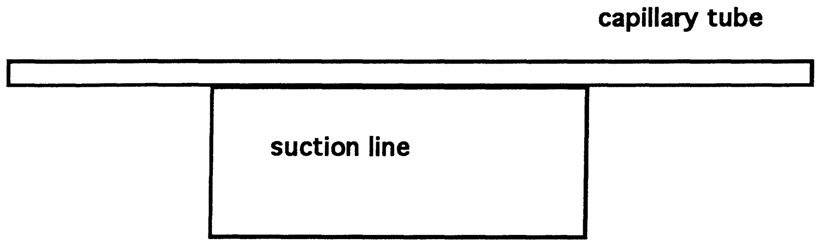
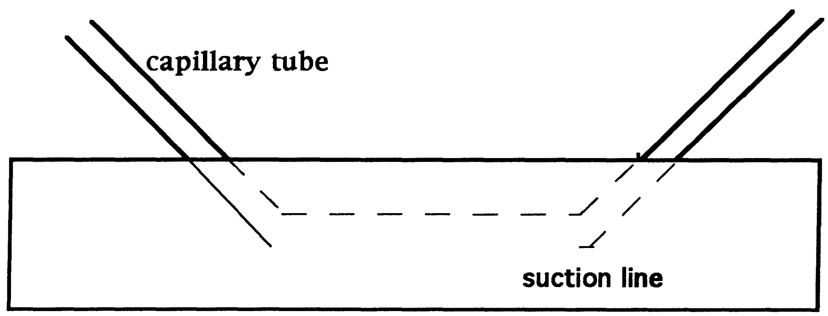


Figure 1. Vapor compression cycle with ct-slhx



a) Lateral Tubes



b) Concentric Tubes

Figure 2. Capillary tube-suction line heat exchanger configurations

Despite its simplicity, the capillary tube-suction line heat exchanger is one of the most difficult components of the system to design. Historically the approach has been almost completely empirical. The ASHRAE Equipment Handbook, for example, presents a chart defining the pressure drop-mass flow relationship for the adiabatic case as a function of inlet conditions. With suction line heat exchange, however, this relationship is altered and the charts yield only approximate values based on the assumption that the refrigerant remains subcooled liquid along the entire length of the heat exchanger. Much laboratory time is required to test the system under a variety of conditions to adjust the dimensions of the ct-slhx.

The design problem has been further complicated by the need to phase out CFCs and HCFCs, which has diminished the value of vast empirical databases and design experience. Although it is likely that HFC-134a will be used in most new refrigerators, there is at present no experimental and theoretical data are available in the open literature for describing the performance of non-adiabatic capillary tube using HFC-134a as a working fluid.

This paper presents a model of the relationship between refrigerant flow rate and pressure drop in adiabatic capillary tubes, and in suction line heat exchangers. The models depend on only a few simplifying assumptions. They can be used for sizing and simulation and account for the properties of alternative refrigerants. The next two sections describe the processes and earlier approaches to modeling it. The next two sections present the our model and compare its results to published data. Finally the design capabilities of the model are illustrated by a set of parametric analyses.

Description of the process

The capillary tube connects the condenser and the evaporator, and the refrigerant can be either liquid subcooled or two-phase at its inlet. The flow through a capillary tube can generally be divided into a liquid region, where the pressure decreases linearly until the flash point; and a two-phase region characterized by increasing refrigerant velocity and pressure drop per unit length as the exit is approached. The following characteristics account for the complexity of the process:

(i) Flashing two-phase flow differs somewhat from from the classical two-phase boiling. In a capillary tube-suction line heat exchanger, the flow is still more complex because of the refrigerant being simultaneously cooled.

(ii) When the refrigerant enters as a subcooled liquid, the pressure decreases steadily to the saturation pressure where the vaporization should begin. During the last 30 years, several authors have observed experimentally the existence of a delay in the refrigerant vaporization, called "metastable region," in adiabatic capillary tubes. It was verified that the temperature remains constant for some distance past the saturation point (A) and the inception of vaporization (B) shown in the Figure 3. The existence of this phenomenon for non-adiabatic capillary tube is much more difficult to detect experimentally. The unique work that was concerned about this problem, (Pate, 1982, 1984a), was not conclusive about the existence of the phenomenon of "metastability."

(iii) The refrigerant vaporization increases specific volume and therefore velocity, and it is common to reach the critical (choked) flow condition at the tube exit. At a fixed condenser pressure, further reductions of the evaporator pressurebelow this point will not increase the mass flow rate.

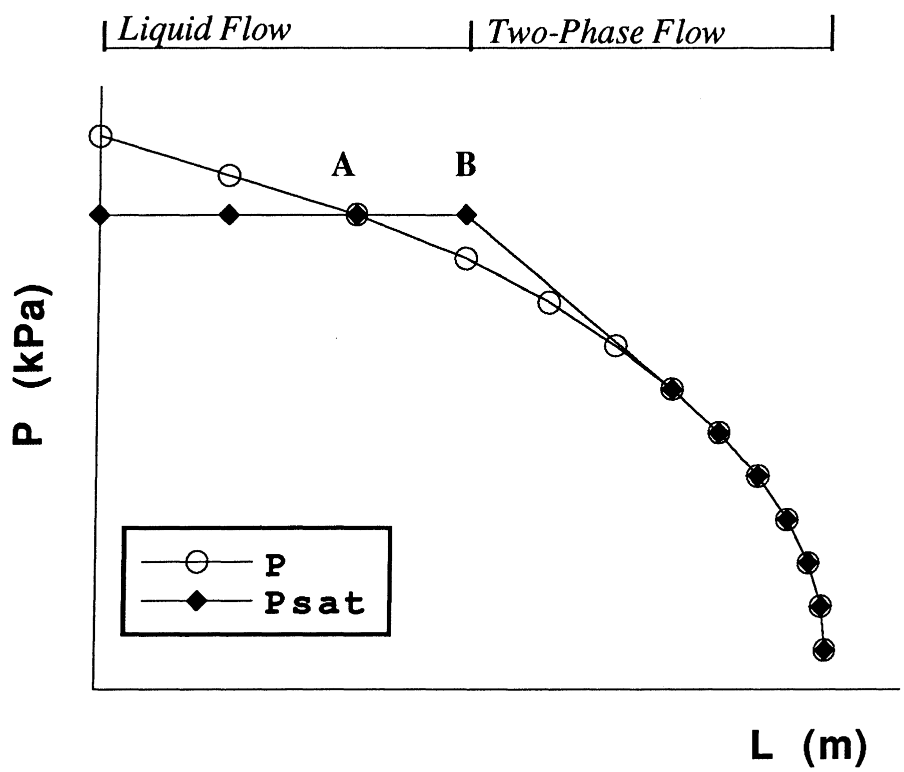


Figure 3. Typical pressure profile for an adiabatic capillary tube

Brief literature review

Most research reported in the open literature deals with adiabatic capillary tubes. Schulz (1985), Kuehl and Goldschmidt (1991) and Purvis (1992) described the important characteristics and conclusions of these works. Fewer studies have been done on non adiabatic capillary tubes; some of them are described below.

Christensen (1967) performed a series of tests with a capillary tube-suction line heat exchanger at varying evaporator and condenser pressures and with the heat exchange region changeable from one end to another of the capillary tube. The length of the heat exchange region was fixed at 1000 mm. Tests were performed at evaporating temperatures in the range -5.0 to -25.0 ^{o}C corresponding to condensing temperatures in the range 30.0 to 50.0 ^{o}C. These tests were conducted for three positions of the heat exchanging section and for each test the refrigerant (CFC-12) mass flow rate was measured. It was verified that the evaporating pressure has little influence while the condensing pressure has a great effect on the refrigerant mass flow rate. Mass flow rates increased as the free length of the capillary tube was moved from the condenser towards the evaporator for fixed condenser and evaporator pressures.

Pate and Tree (1983, 1984a, 1984b, 1987) published several papers based in the Pate's Ph.D. research (1982) about the flow of CFC-12 though capillary tube-suction line heat exchanger. These papers presented described the experimental apparatus used, the results of the tests performed, a mathematical model for simulation, and a study of the choked flow conditions at the capillary tube outlet. Model results for length calculations showed fair agreement with the experimental data. However to simplify the computational procedure, the authors assumed a linear profile of the quality in the heat exchanger region. As pointed out by Yan and Wang (1991) this assumption simplified the calculation but the actual refrigerant quality distribution in the capillary tube depends on the rate of heat exchange with the suction line.

The interaction between the capillary tube-suction line heat exchanger and the whole system was studied by Pereira et al (1987). They analyzed experimentally the thermal performance of a domestic refrigerator, using CFC-12 as working fluid, as the heat exchanging conditions at the capillary tube and the suction line were modified. They documented the improvement in overall performance of the refrigerator and verified that the effect of the heat exchange with the surrounding air was negligible.

The ct-slhx simulation presented by Q.Yan and X.L.Wang (1991) was based on the fundamental fluid dynamics and heat transfer equations. However some of the correlations for heat transfer coefficient and friction factor used are unfamiliar and the paper contains insufficient details about them. The paper presented results (mass flow rate, quality profile, etc.) for CFC-12 for a fixed geometry over a range of inlet and outlet pressures and inlet subcooling. They showed that small changes in mass flow can produce large changes in the quality along the slhx length. They suggest that it might be related to the metastable phenomenon that can occur as described in ASHRAE (1988).

Domanski et al.(1992) presented a theoretical evaluation of the performance effects resulting from the addition of a liquid line-suction line heat exchanger to a standard vapor compression refrigeration cycle. It demonstrated how installation of a liquid line-suction line heat exchanger improves COP and volumetric capacity in the case of fluids that perform poorly in the basic cycle. The benefits obtained depends on the combination of the operating conditions and fluid properties with the liquid and vapor specific heats being the most important. The improvement presented by HFC-134a is 2.5% higher than for CFC-12. The results presented, as noted by the author, are for ideal cases and don't consider some factors that are present in the real systems and affect their performance. For household refrigerators the heat exchange is not between the liquid line and the suction line but between the capillary tube, where the refrigerant is flashing, and the suction line. The usual approach of idealizing the ct-slhx as an isobaric nonadiabatic section followed by an isenthalpic expansion is a simplification of the real process (A. D. Little, 1982 and ASHRAE, 1988).

Proposed capillary tube model

Over the last fifty years capillary tube modeling efforts have developed as follows:

- graphical integration, for adiabatic flow, of momentum or momentum and energy equations using average thermodynamic and transport properties and friction factor;
- numerical integration, for adiabatic flow, of momentum and energy equations considering homogeneous equilibrium model for two-phase flow, and local calculation of thermodynamic and transport properties and friction factor;
- modeling non adiabatic flow considering the heat exchange with suction line through simultaneous integration of momentum and energy equation for capillary tube with energy equation for suc.line and heat exchanger;
- use, for adiabatic flow, of nonequilibrium two-phase flow model which account for the thermal and hydrodynamic non equilibrium between the phases

The approach taken here builds on most of the assumptions and correlations that have been verified in the literature, and employs a Newton-Raphson solution technique that makes it unnecessary to assume a linear quality profile. The results are then compared to published experimental data for various refrigerants. Then simulations are performed to aid in designing a test matrix for subjecting the assumptions to further experimental validation to determine systematically which assumptions might have to be relaxed. For adiabatic flow it is assumed that:

a) negligible heat exchange with the ambient;
b) steady state, pure refrigerant one-dimensional flow;
c) homogeneous equilibrium two-phase flow;
d) critical conditions reached when Mach number of the homogeneous liquid and vapor mixture at the exit section is equal 1.0.

The homogeneous two-phase model assumes a thermal and hydrodynamic equilibrium between the phases (equal temperature and velocities; no delay of vaporization) and provides good results when there is sufficient time for the two phases to reach equilibrium as might occur in long tubes (Dobran, 1987).

The additional assumptions for the heat exchanger region, in the case of capillary tube-suction line heat exchanger, are:

e) negligible axial heat conduction in the capillary tube and suction line walls;
f) negligible thermal resistance in the capillary tube and suction line walls,
g) radially and axisymetrically isothermal capillary tube and suction line walls.

The figures 4 and 5 define the variables used in the model for adiabatic capillary tube and the capillary tube-suction line heat exchanger composed of lateral tubes. The variables for the concentric tubes capillary tube-suction line heat exchanger are the same, with the addition of the capillary tube external diameter.

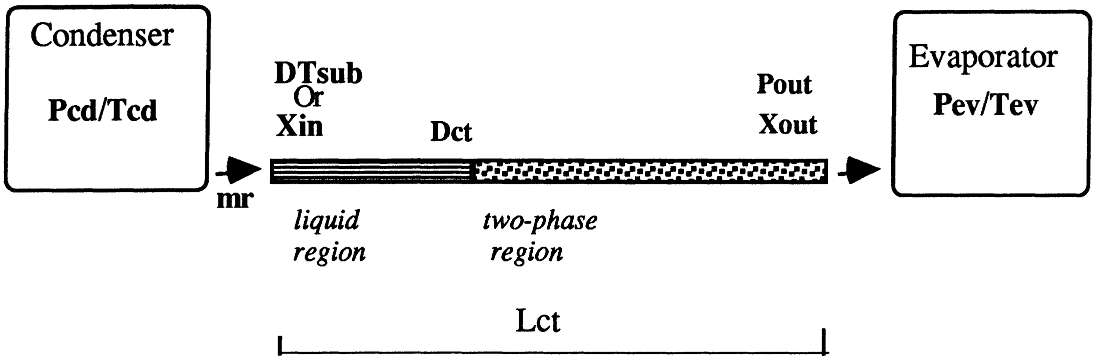


Figure 4. Variables used in the adiabatic capillary tube model.

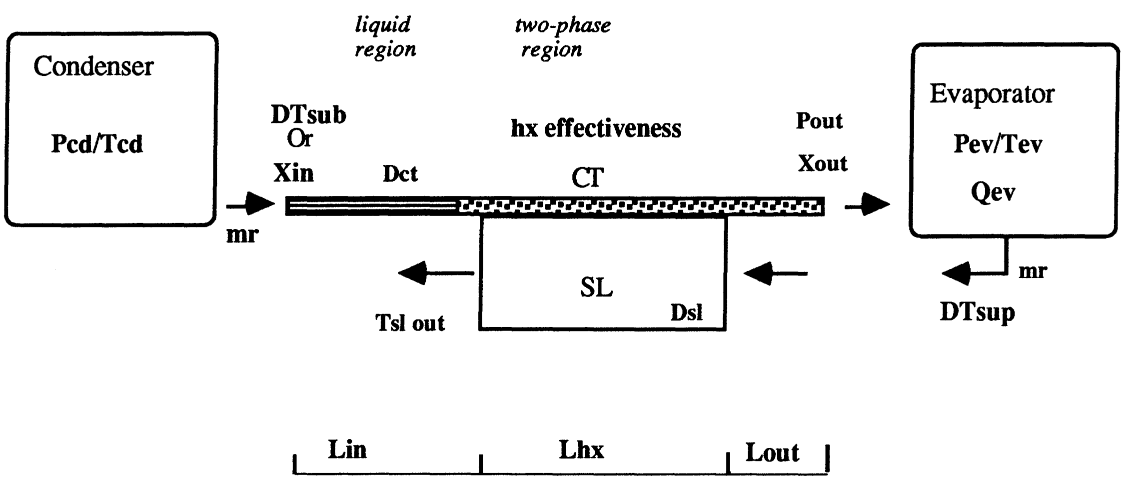


Figure 5. Variables used in the capillary tube-suction line heat exchanger model

The flashing point in Figure 5 lies in the adiabatic inlet region, but it can be located also in the heat exchange and in the adiabatic outlet region.

The governing equations are the mass, momentum and energy conservation equations, presented bellow:

For adiabatic region:

$$\frac{\dot{m}_r}{A_{ct}} = G_{ct} = const \qquad (1)$$

$$\frac{-dp}{dx} = \frac{f \upsilon G_{ct}^{\ 2}}{2D_{ct}} + G_{ct}^{\ 2}\frac{d\upsilon}{dx} \qquad (2)$$

$$\frac{di}{dx} = \frac{G_{ct}^{\ 2}}{2}\frac{d\left(\upsilon^2\right)}{dx} \qquad (3)$$

For the heat exchanger region, considering the lateral design:

$$\frac{\dot{m}_r}{A_{ct}} = G_{ct} = const.$$

$$\frac{\dot{m}_r}{A_{sl}} = G_{sl} = const. \tag{4}$$

$$\frac{-dp}{dx} = \frac{f\upsilon G_{ct}{}^2}{2D_{ct}} + G_{ct}{}^2 \frac{d\upsilon}{dx}$$

$$\frac{di}{dx} + \frac{G_{ct}{}^2}{2}\frac{d\left(\upsilon^2\right)}{dx} = cp_{sl}\frac{dT_{sl}}{dx} \tag{5}$$

$$\dot{m}_r cp_{sl}\frac{dT_{sl}}{dx} = -h_{sl}\pi D_{sl}\left(T_w - T_{sl}\right) \tag{6}$$

$$h_{ct}\pi D_{ct}\left(T_{ct} - T_w\right) - h_{sl}\pi D_{sl}\left(T_w - T_{sl}\right) = 0 \tag{7}$$

For the heat exchanger region, considering concentric tubes:

$$\frac{\dot{m}_r}{A_{ct}} = G_{ct} = const.$$

$$\frac{\dot{m}_r}{\frac{\pi(D_{sl}^2 - OD_{ct}^2)}{4}} = G_{sl} = const. \tag{8}$$

$$\frac{-dp}{dx} = \frac{f\upsilon G_{ct}{}^2}{2D_{ct}} + G_{ct}{}^2 \frac{d\upsilon}{dx}$$

$$\frac{di}{dx} + \frac{G_{ct}{}^2}{2}\frac{d\left(\upsilon^2\right)}{dx} = cp_{sl}\frac{dT_{sl}}{dx}$$

$$\dot{m}_r cp_{sl}\frac{dT_{sl}}{dx} = -h_{sl}\pi OD_{ct}\left(T_w - T_{sl}\right) \tag{9}$$

$$h_{ct}\pi D_{ct}\left(T_{ct} - T_w\right) - h_{sl}\pi OD_{ct}\left(T_w - T_{sl}\right) = 0 \tag{10}$$

The constitutive equations are:

$$Dp = (1+K)\frac{V_{in}{}^2}{2\upsilon_{in}} \tag{11}$$

where Dp is pressure drop between the condenser pressure and the inlet pressure due to the joining of the capillary tube with the liquid line. For the two-phase region the thermodynamics and transport properties are:

$$h=(1-x)h_f+xh_g \tag{12}$$

$$s=(1-x)s_f+xs_g \tag{13}$$

$$\upsilon=(1-x)\upsilon_f+x\upsilon_g \tag{14}$$

$$\kappa=(1-x)\kappa_f+x\kappa_g \tag{15}$$

$$c_p=(1-x)c_{p_f}+xc_{p_g} \tag{16}$$

$$\mu=\frac{(X\upsilon_g\mu_g+(1-X)\upsilon_f\mu_f)}{\upsilon} \tag{17}$$

The thermodynamic and the transport properties for the liquid and vapor phases are calculated using Martin Hou equation of state and the transport property correlations presented by Shankland et al. (1989) and Jung and Radermacher (1991). The two-phase viscosity is determined through the correlation proposed by Dukler (1964). The friction factors for the liquid region are calculated using Blasius equation for turbulent flow in smooth tubes, and in the two-phase region using a experimental correlation obtained by Pate (1982). It is expected that the latter may need to be modified for other refrigerants.

$$f_{liq}=0.3164\,\mathrm{Re}^{-0.25} \tag{18}$$

$$f=3.49\,\mathrm{Re}^{-0.47} \tag{19}$$

Single-phase heat transfer coefficients are calculated through the Dittus Boelter equation:

$$Nu=0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.3} \qquad \text{(cap. tube)} \tag{20}$$

$$Nu=0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \qquad \text{(suction Line)} \tag{21}$$

and the sonic velocity and the refrigerant velocity are calculated by the relations:

$$G_{ct}=\frac{V}{\upsilon} \tag{22}$$

$$c^2=\left(\frac{\partial p}{\partial \rho}\right)_S \tag{23}$$

Validation

The model was validated by comparing with published data for both the adiabatic capillary tube and for the capillary tube-suction line heat exchanger.

The finite difference method was used to solve the governing differential equations, continuity, momentum and energy, in conjunction with the constitutive equations. The resulting system of algebraic equations are solved by the Newton-Raphson method. The mass flow rate is calculated for a set of operational conditions (inlet pressure, inlet subcooling or quality, etc.) and geometry (lengths and internal diameters). For the simulation problem in which mass flow is unknown, the equations must be solved simultaneously. On the other hand for the design problem in which the mass flow rate is fixed, the solution for length is nearly sequential and the problem of initial guesses is minimized. The solution also yields quality, pressure, temperature and enthalpy distributions.

For the adiabatic case figures 6-8 compare the calculated mass flow with experimental data for the refrigerants HCFC-22, CFC-12 and HFC-134a. Also shown are results obtained using the ASHRAE (1988) procedure, which underpredicted the mass flow rate. The mass flows predicted by the model are consistently closer to the actual values reported by Kuehl and Goldschmidt (1991) and Wijaya (1991) for both CFC-12 and HCFC-22.

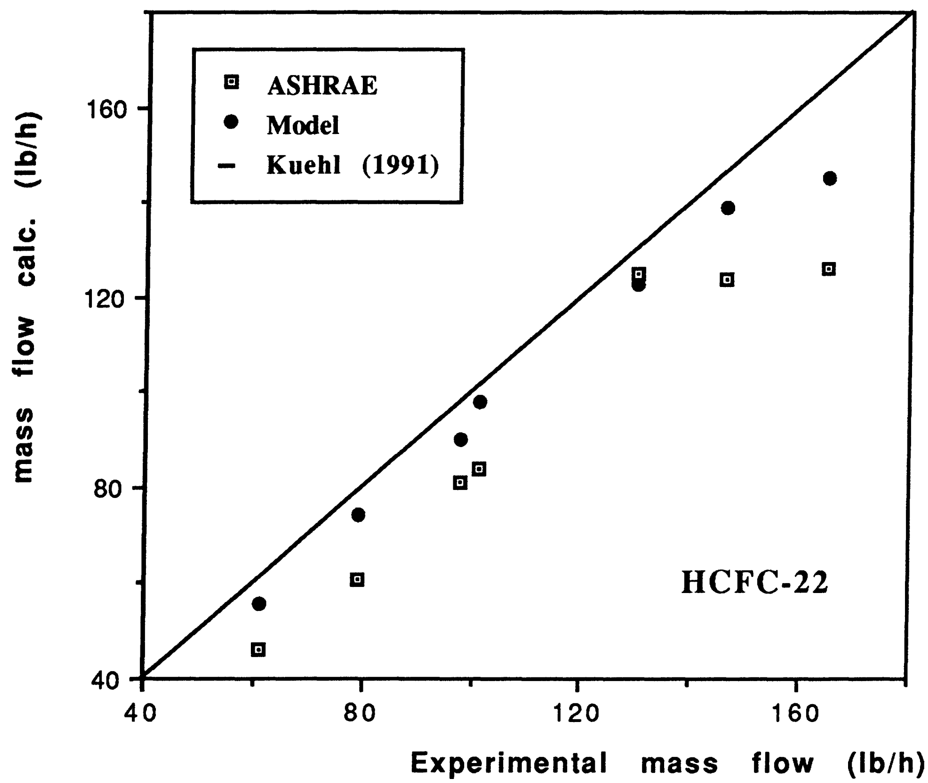


Figure 6. Validation of adiabatic capillary-tube model for HCFC-22

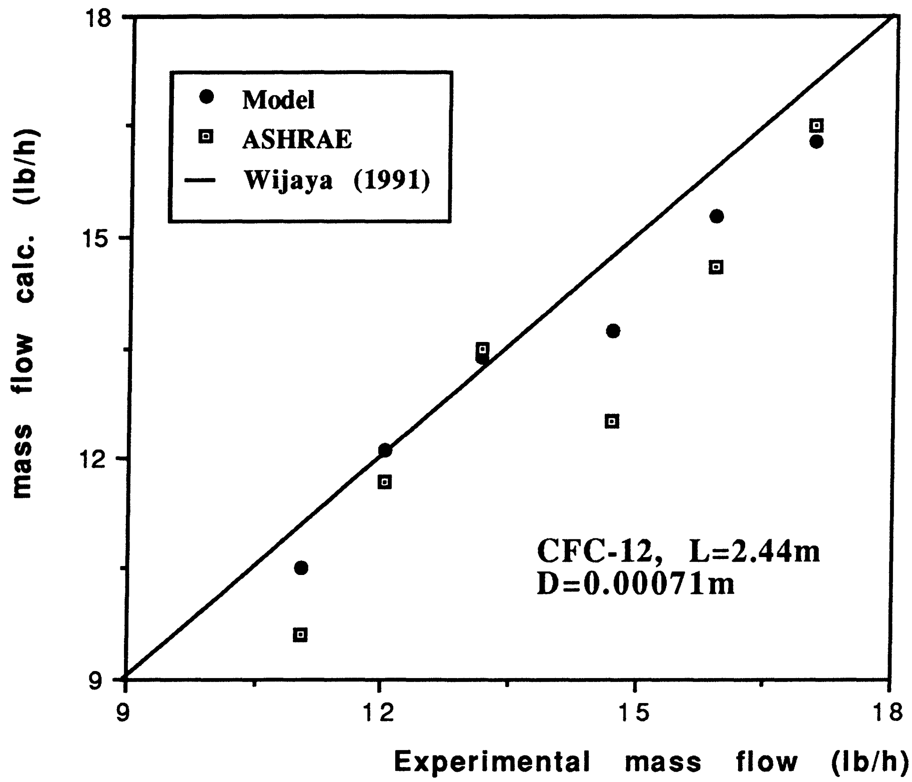


Figure 7. Validation of adiabatic capillary tube model for CFC-12.

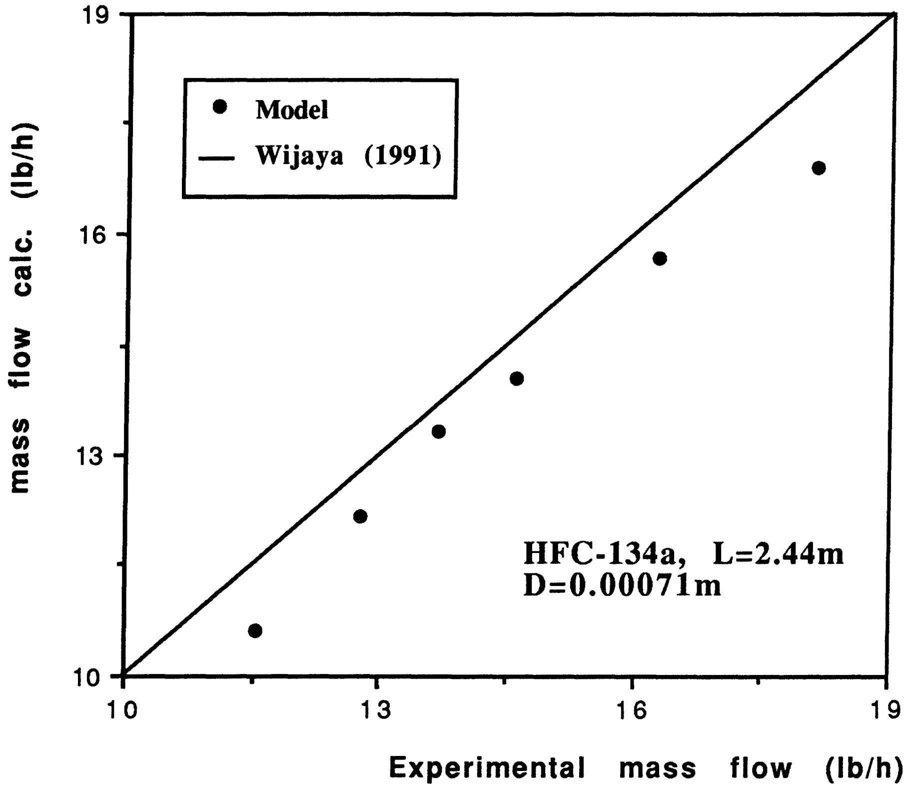


Figure 8. Validation of adiabatic capillary tube model for HFC-134a.

Next the model was run for a wide range of conditions in order to develop equations for the mass flow rate that could be integrated into a system simulation model without the complexity and convergence problems of the finite-difference method. The results were used to fit equations of the form:

$$\dot{m}_r = \mathrm{GCF}\,\dot{m}_{std} \tag{24}$$

$$\dot{m}_{std} = a_1 P_{in} + a_2 DT_{sub} + a_3 P_{in}^{\ 2} + a_4 DT_{sub}^{\ 2} + a_5 \tag{25}$$

$$\mathrm{GCF} = a_6 \left(\frac{D}{D_{std}}\right)^{a_7} \left(\frac{L}{L_{std}}\right)^{a_8} + a_9 \tag{26}$$

where GCF is the geometric geometry correction factor similar to the flow factor used in ASHRAE charts (1988).

Very little experimental data have been published for diabatic case of a capillary tube-suction line heat exchanger. Figures 9 and 10 show good agreement between the mass flow rate calculated by the model and the data presented by Christensen (1967) for CFC-12. However it must be noted that these data are for a high degree of subcooling at the capillary tube inlet. In this condition, the liquid region is large and the liquid flow is well predicted by the theory. The predicted flashing point was located near the end of the heat exchange region or in the adiabatic outlet region, far downstream of its usual position. More theoretical and experimental work is needed to validate the model in the two-phase flow region. This research is currently underway and the results will be reported in the future.

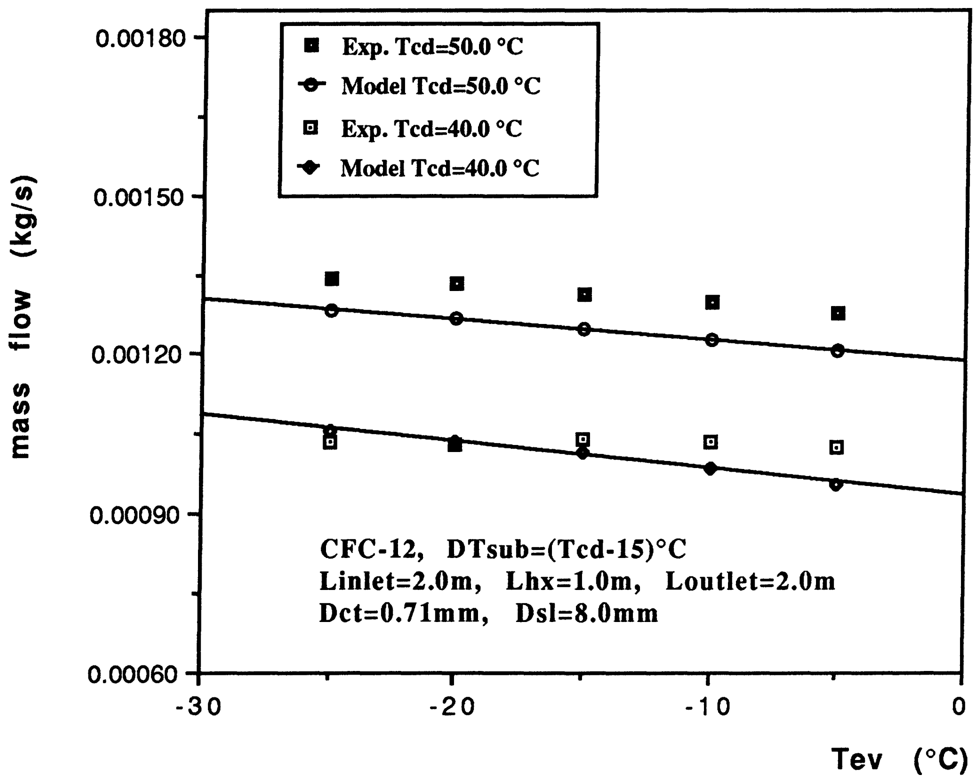


Figure 9. Validation of ct-slhx model (long entrance section)

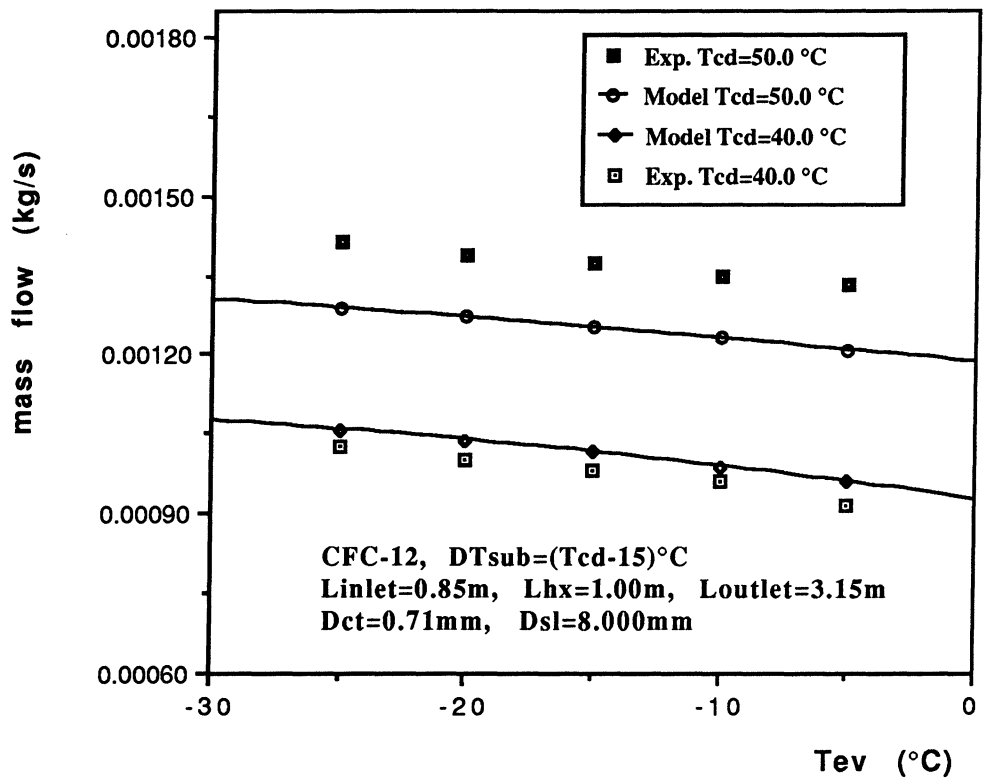


Figure 10. Validation of ct-slhx model (short entrance section)

Simulation and design results

Figures 11 and 12 illustrate the use of the capillary tube-suction line heat exchanger model for simulation. Figure 11 suggests that HFC-134a will respond similarly to CFC-12 as condensing temperature changes. Figure 12 shows that the mass flow rate of HFC-134a is expected to be more sensitive to the modest amounts of subcooling typical of refrigerator operating conditions.

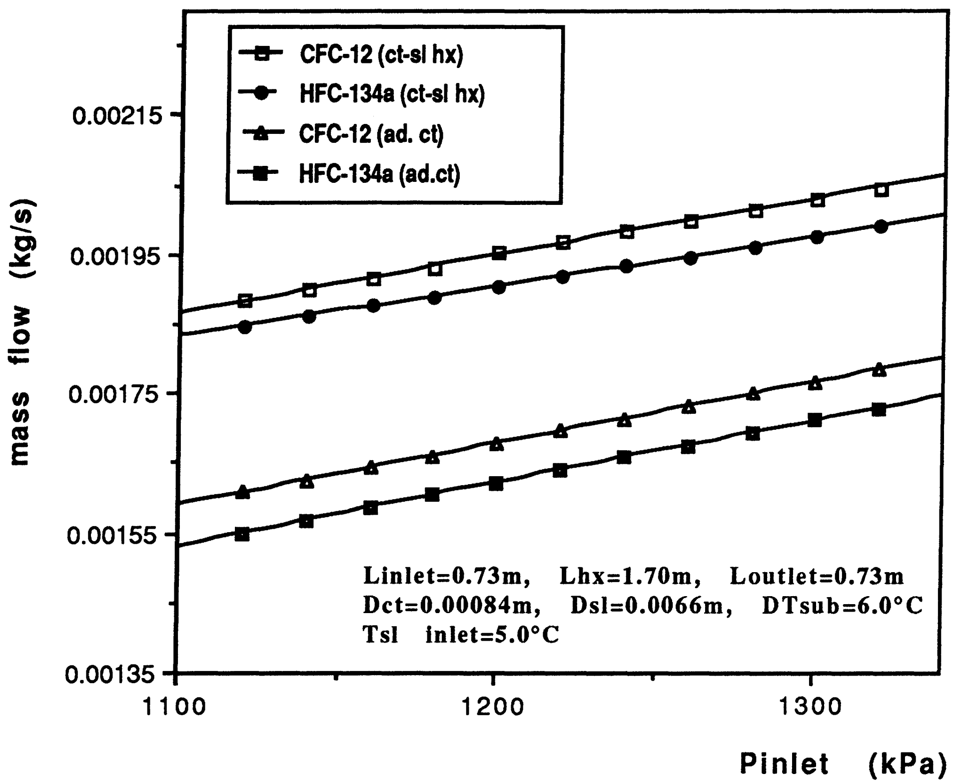


Figure 11. Effect of the condenser pressure on mass flow

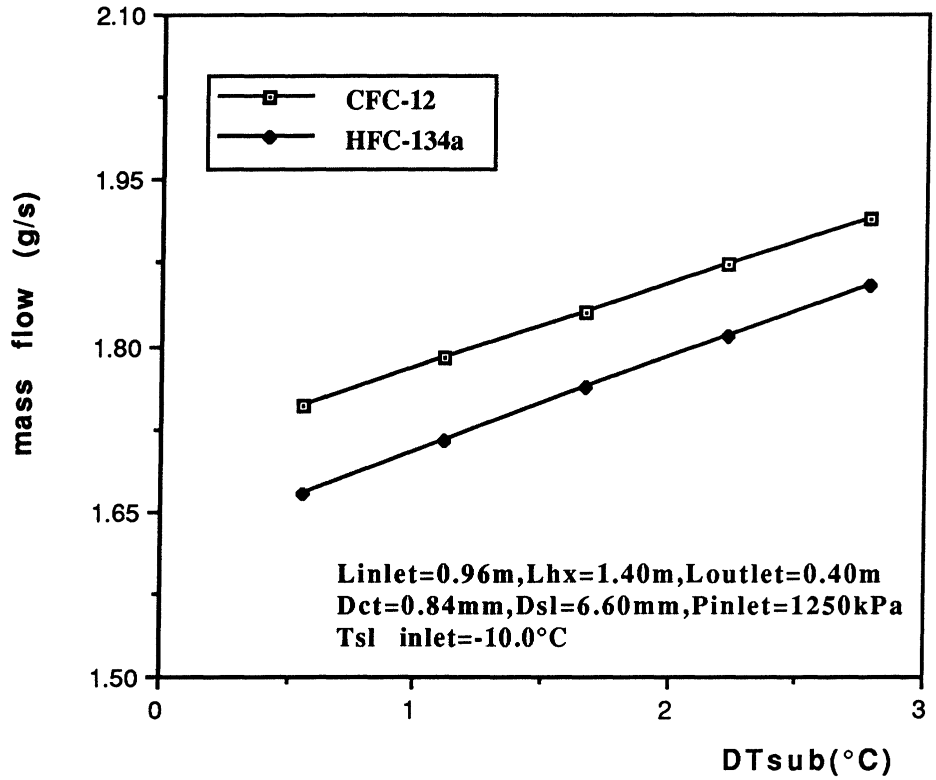


Figure 12. Effect of the inlet subcooling on mass flow simulation

The model can also be used for designing suction line heat exchangers and examining tradeoffs between the various tube lengths, diameters, and the way they are assembled. Figure 13 shows the importance of the suction line diameter, which defines the heat transfer area on the limiting (vapor) side of the heat exchanger. The differences in thermodynamic and transport properties of CFC-12 and HFC-134a are also reflected in this plot. The length of the heat exchanger is taken as the independent variable in Figure 14 to demonstrate its linear dependence on, and sensitivity to, the design condensing pressure. Both figures show results for a specified refrigerant mass flow rate, which is a logical starting point for designing a ct-sl heat exchanger that will exactly match a given compressor capacity at the design condition.

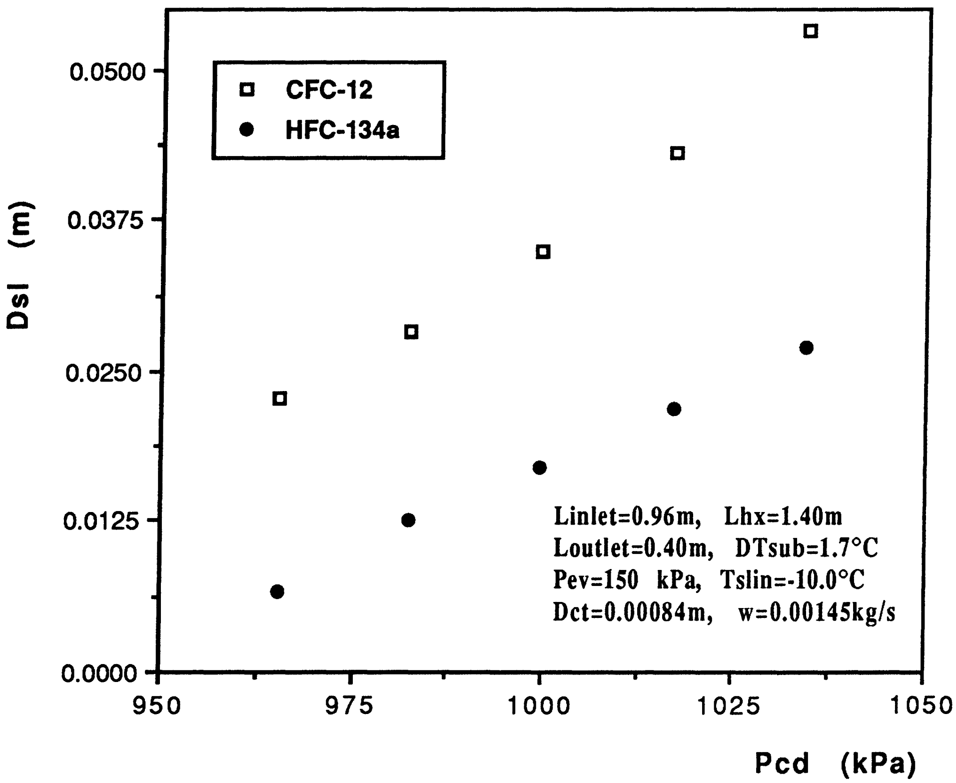


Figure 13. Effect of the condenser pressure in the suction line diameter calculation (lateral ct-sl hx)

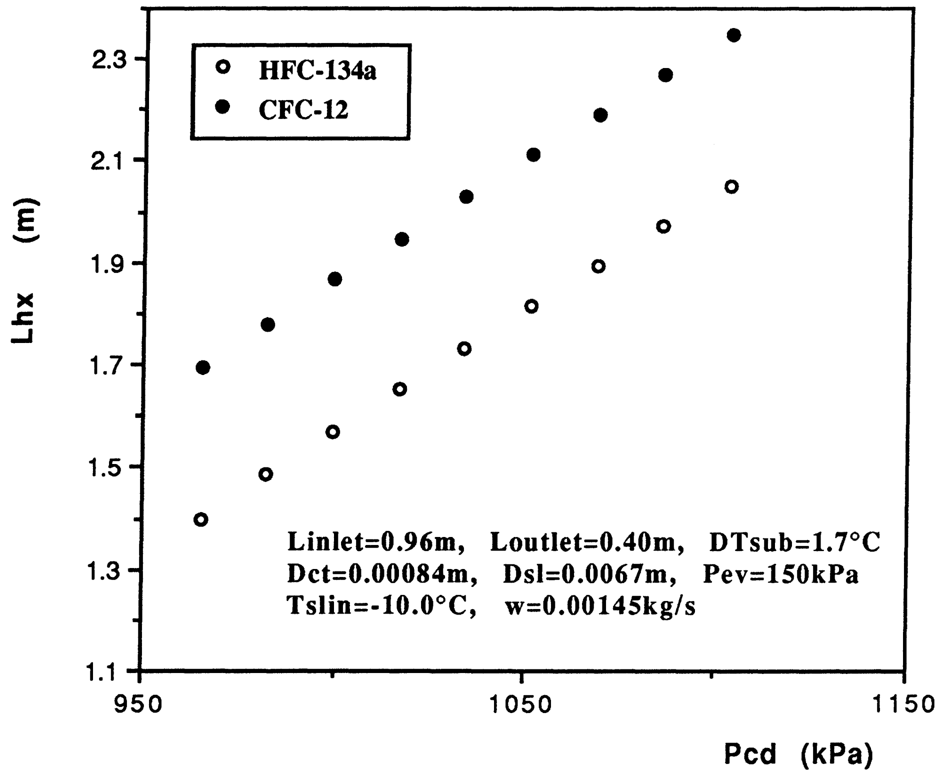


Figure 14. Effect of the condenser pressure in the heat exchange length calculation (lateral ct-sl hx)

Another important design consideration is the potential for recondensation in the heat exchanger portion of the capillary tube, as illustrated by the nonlinear quality profile in Figure 15. The ASHRAE handbook cautions that recondensation can cause instabilities in refrigerator operation. Avoiding this condition may place a practical upper limit on the effectiveness of the heat exchanger. This simulation result shows clearly that a linear quality profile is unlikely to exist in most cases.

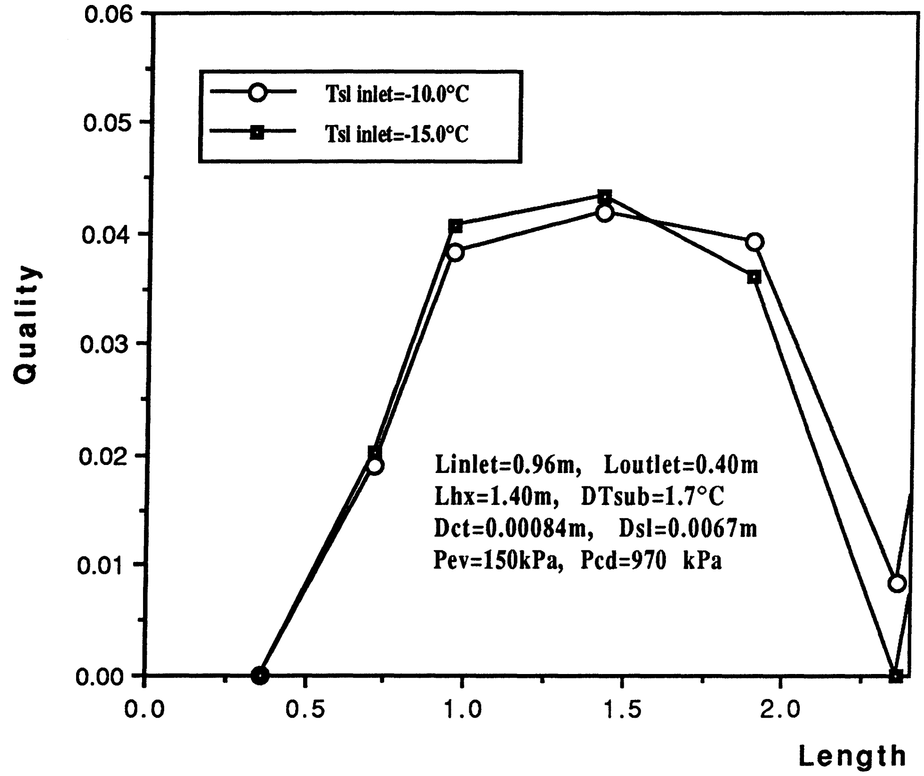


Figure 15. Quality profile in the capillary tube

Finally Figures 16 and 17 illustrate important differences between lateral and concentric designs. The calculations suggest that the superior performance of the lateral design results from the capillary tube's ability to reject heat via conduction to the larger suction line, instead of presenting its small outside surface area to the suction gas.

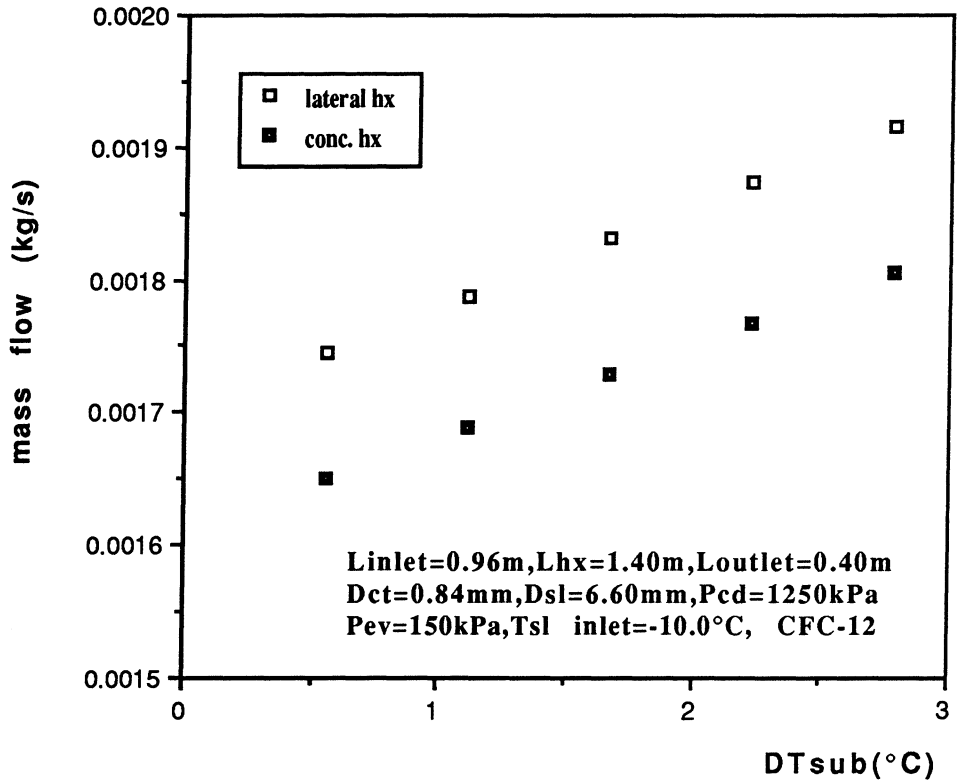


Figure 16. Mass flow calculation for lateral and concentric tubes ct-sl hx

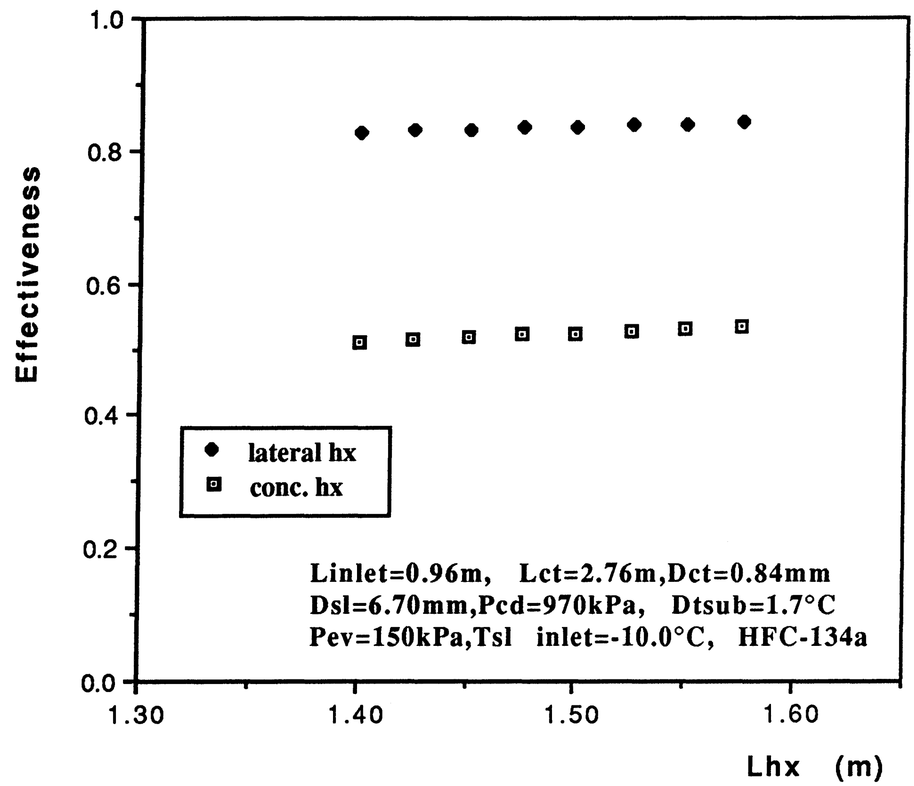


Figure 17. Effectiveness calculation for lateral and concentric tubes ct-sl hx

Conclusions

The finite difference model presented here shows reasonably good agreement with published data for both adiabatic and diabatic capillary tubes. The agreement is better than obtainable with the widely-used charts published in the ASHRAE handbook for CFC-12 and HCFC-22. Moreover, this model can be used to predict the performance of alternative refrigerants. Much work needs to be done in order to validate the two-phase flow equations and to analyze the problem of refrigerant re-condensation in the capillary tube near the end of the heat exchange region. To help develop a test matrix for the required validation experiments, the model was run in both design and simulation mode for a wide range of operating conditions. Important trends are summarized here.

The mass flow rate of HFC-134a is expected to be about 5% lower than CFC-12 in an adiabatic capillary tube. Adding a capillary tube-suction line heat exchanger increases mass flow about 20% for both refrigerants. For a given mass flow rate the calculations suggest that HFC-134a will require substantially a smaller suction line diameter and heat exchange length. Finally the model illustrates how the effectiveness of a capillary tube-suction line heat exchanger is higher for the case of lateral tubes than for concentric tubes, because of the larger heat transfer area presented by isothermal capillary tube and suction line walls.

References

American Society of Heating, Refrigerating and Air-Conditioning Engineers, 1988. ASHRAE Handbook, Equipment Volume, Atlanta, GA.

Arthur D. Little, Inc., *Refrigerator and Freezer Computer Model User's Guide*, U. S. Department of Energy DE-AC01-78CS20420, November 1982.

Christensen, L. B. and Jorgensen, H. P., "Flow Resistance of a Capillary Tube Heat Exchanger," Proc. XII Int. Congr. of Refrigeration, pp 1069-1077, Madrid, Spain, 1967.

Dobran, F., "Nonequilibrium Modelling of Two-Phase Critical Flow in Tubes," Journal of Heat Transfer, vol. 109, pp. 731-738, 1987.

Domanski P. A., Didion D. A., and Doyle J. P., 1992, Evaluation of Suction Line-Liquid Line Heat Exchange in the Refrigeration Cycle. in Proceedings of the International Refrigeration Conference, Purdue University, West Lafayette, IN.

Dukler, A.E. et al., "Pressure Drop and Hold Up in Two-Phase Flow" A. I. Ch. Journal, vol. 10, pp 38-51, 1964.

Jung, D. and Radermacher, R., 1991, "Transport properties and surface tension of pure and mixed refrigerants," ASHRAE Transactions, **97** (Pt. 1).

Kuehl, S. J., and Goldschmidt, V. W., 1991, "Modeling of steady flows of R-22 through capillary tubes." ASHRAE Transactions, **96**, Part 1, pp. 139-148.

Pate, M. B., 1982, "A Theoretical and Experimental Analysis of Capillary Tube-Suction Line Heat Exchangers," Ph.D., Thesis, Purdue University.

Pate, M. B. and Tree, D. R., 1983, "An Experimental Analysis of a Capillary Tube-Suction Line Heat Exchanger," Proceedings of XVIth International Congress of Refrigeration 1983, 162-167, Paris, France.

Pate, M. B. and Tree, D. R., 1984, "A Linear Quality Model for Capillary Tube-Suction Line Healt Exchangers," ASHRAE Transactions, **90** (Pt. 2a), 3-17.

Pate, M. B. and Tree, D. R., 1984, "An Analysis of Pressure and Temperature Measurements Along a Capillary Tube-Suction Line Heat Exchanger," ASHRAE Transactions, **90** (Pt. 2a), 291-301.

Pate, M. B. and Tree, D. R., 1987, "An Analysis of Choked Flow Conditions in a Capillary Tube-Suction Line Heat Exchanger," ASHRAE Transactions, **93** (pt. 1), 368-380.

Pereira, R. H. et al, "Effect of capillary tube-suction line heat exchanger in a household refrigerator performance, (in portuguese) Proc. of the 9th Brazilian Congress of Mech. Engineering, pp 355-357, Sao Paulo, SP Brazil.

Schulz, U. W., 1985, "State of the Art: The Capillary Tube for, and in, Vapor Compression Systems," ASHRAE Transactions, **91** (Pt. 1b), 92-103.

Schulz, U. W., 1988, "A Correlation of Experimental Exit Pressures for Refrigerant Flow Through a Capillary Tube Expansion Device," ASHRAE Transactions, **94** (Pt. 1), 507-519.

Shankland, I. R, Basu, R. S., and Wilson, D. P., "Thermal Conductivity and Viscosity of a New Stratospherically Sate Refrigerant - 1,1,1,2 Tetrafluoroethane (R-134a)," published in **CFCs: Time of Transition**, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., 1989.

Wijaya H., 1991, "An Experimental Evaluation of Adiabatic Capillary Tube Performance for HFC-134a and CFC-12," Proceedings of the International CFC and Halon Alternatives Conference at Baltimore, MD.

Yan, Q. and Wang, X. L., 1991, "Studies on the Behavior of a Capillary Tube in a Small Refrigeration System," Presented at the International Congress of Refrigeration at Montreal, Canada.