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A SIMULATION AND DESIGN MODEL FOR CAPILLARY TUBE-SUCTION LINE HEAT EXCHANGERS

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ABSTRACT

This paper describes a model for capillary tube-suction line heat exchangers that can be used for either simulation or design. This model can also be integrated into a refrigeration system simulation and is valid for alternative refrigerants. The model permits, for a specific refrigerant, the choice of three dependent variables. For a given set of operational and geometric conditions the model calculates mass flow rate, capillary tube exit quality and suction line temperature at heat exchanger outlet. Alternatively in design mode, the model can determine the lengths and tube diameters required for a given flow rate and operational conditions. Results are presented for CFC-12 and HFC-134a.

NOMENCLATURE

A = cross sectional area (m^2), (ft^2)
c = sound velocity (m/s), (ft/s)
cd = condenser
c_p = const. pressure heat capacity (J/kg °C), (Btu/lb °F)
D = internal diameter (m), (ft)
$DTsub$ = subcooling (°C), (°F)
$DTsup$ = superheating (°C), (°F)
ev = evaporator
f = friction factor
G = mass flux (kg/s m^2), (lb/h ft^2)
h = convective heat transfer coef. (W/m^2 °C), (Btu/h.ft^2 °F)
h = specific enthalpy (J/kg), (Btu/lb)
hx = heat exchanger
k = thermal conductivity (W/m °C), (Btu/h ft °F)
K = entrance loss factor
L = length (m), (ft)
$\dot{m}$ = mass flow rate (kg/s), (lb/h)
Nu = Nusselt number
OD = external diameter (m), (ft)
p = pressure (kPa), (psi)
Pr = Prandtl number
Re = Reynolds number
s = specific entropy (J/kg °C), (Btu/lb °F)
T = temperature (°C), (°F)
v = specific volume (m^3/kg), (ft^3/lb)
V = velocity (m/s), (ft/s)
x = quality
z = distance (m), (ft)
μ = viscosity (Pa.s), (lbf s/ft^2)
ρ = density (kg/ m^3), (lb/ ft^3)

Subscripts

ct = capillary tube
l = liquid phase
v = vapor phase
i = inner
in = inlet
out = outlet
sl = suction line
w = tube wall

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 (called concentric tubes ct-sl hx, regardless of the relative location of the tubes).

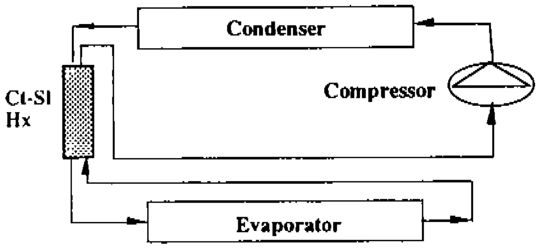

Figure 1. Vapor compression cycle with ct-slhx

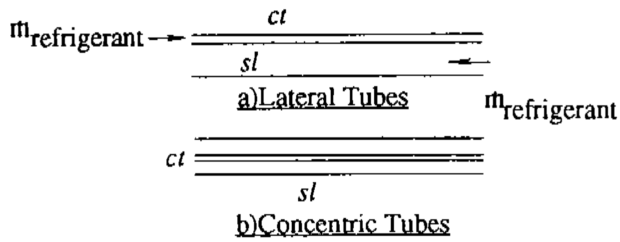

Figure 2. Capillary tube-suction line heat exchanger

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 available in the open literature for describing the performance of non-adiabatic capillary tube using HFC-134a as a working fluid.

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 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 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 and Tree 1984), 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 pressure below this point will not increase the mass flow rate.

PROPOSED MODEL

In almost all the applications just the intermediate part of the capillary tube is placed in contact with suction line. This is simulated in the model by dividing the capillary tube into three regions, two adiabatic and one non adiabatic. For adiabatic flow it is assumed that:

> a) negligible heat exchange with the ambient; b) steady state, pure refrigerant one-dimensional flow; c) fully developed turbulent flow; d) homogeneous equilibrium two-phase flow; e) 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 are:

f) fully developed turbulent flow in suction line; g) negligible axial heat conduction in the capillary tube and suction line walls; h) negligible thermal resistance in the capillary tube and suction line walls, i) negligible thermal resistance in the soldered joint; j) radially and axisymetrically isothermal capillary tube and suction line walls (lateral ct-sl hx); k) capillary tube and suction line placed in a concentric way (concentric ct-sl hx).

Figure 3 defines the variables used in the model for 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. The flashing point in Figure 3 lies in the adiabatic inlet region, but it can be located also in the heat exchange and in the adiabatic outlet region.

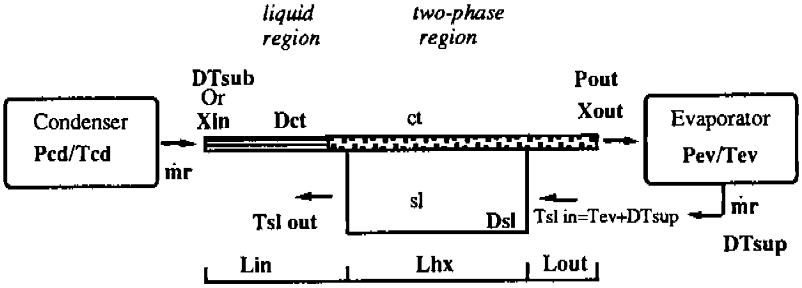

Figure 3. Variables used in the capillary tube-suction line heat exchanger model

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

For adiabatic region:

$$\frac{\dot{m}}{A_{ct}} = G_{ct} = const \quad (1) \qquad \frac{-dp}{dz} = \frac{f\upsilon G_{ct}^2}{2D_{ct}} + G_{ct}^2\frac{d\upsilon}{dz} \quad (2) \qquad \frac{dh}{dz} = \frac{G_{ct}^2}{2}\frac{d(\upsilon^2)}{dz} \quad (3)$$

For the heat exchanger region, considering the lateral design: Equations (1) and (2) above plus:

$$\frac{\dot{m}}{A_{sl}} = G_{sl} = const. \quad (4)$$

$$\frac{dh}{dz} + \frac{G_{ct}^2}{2}\frac{d(\upsilon^2)}{dz} = c_{p_{sl}}\frac{dT_{sl}}{dz} \quad (5)$$

$$\dot{m}c_{p_{sl}}\frac{dT_{sl}}{dz} = -h_{sl}\pi D_{sl}(T_w - T_{sl}) \quad (6)$$

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

For the heat exchanger region, considering concentric tubes: Equations (1), (2) and (5) above plus:

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

$$\dot{m}c_{p_{sl}}\frac{dT_{sl}}{dz} = -h_{sl_i}\pi OD_{ct}(T_w - T_{sl}) \quad (9)$$

$$h_{ct}\pi D_{ct}(T_{ct} - T_w) - h_{sl_i}\pi OD_{ct}(T_w - T_{sl}) = 0 \quad (10)$$

The constitutive equations are:

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

$$h = (1-x)h_l + xh_v \quad (12)$$

$$s = (1-x)s_l + xs_v \quad (13)$$

$$\upsilon = (1-x)\upsilon_l + x\upsilon_v \quad (14)$$

$$\kappa = (1-x)\kappa_l + x\kappa_v \quad (15)$$

$$c_p = (1-x)c_{p_l} + xc_{p_v} \quad (16)$$

$$\mu = \frac{(x\upsilon_v\mu_v + (1-x)\upsilon_l\mu_l)}{\upsilon} \quad (17)$$

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.

The thermodynamic and the transport properties for the liquid and vapor phases are calculated using Martin-Hou equation of state and the transport property equations 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 and two-phase regions are calculated either using Colebrook equation for turbulent flow in rough tubes (if roughness is known), or using an experimental correlation obtained by Pate (1982) :

$$\frac{1}{\sqrt{f}} = -0.86\left(\frac{\varepsilon / D}{3.7} + \frac{2.51}{\mathrm{Re}\sqrt{f}}\right) \quad (18) \qquad\qquad f = 3.49\,\mathrm{Re}^{-0.47} \quad (19)$$

Heat transfer coefficients are calculated through Sleicher and Rouse (1975) correlation:

$$Nu = 5 + 0.015\,\mathrm{Re}^{a}\,\mathrm{Pr}^{b} \quad (20) \qquad a = 0.88 - 0.24 / (4 + \mathrm{Pr}) \quad (21) \qquad b = 0.333 + 0.5e^{-0.6\,\mathrm{Pr}} \quad (22)$$

The concentric ct-sl hx involves heat transfer in a *concentric tube annulus* , and for fully developed turbulent flow, the inner convection coefficient (h_{sl_i}) may be evaluated by using the hydraulic diameter (D_{sl}-OD_{ct}) with the Sleicher an Rouse equation (Incropera and De Witt, 1990). The refrigerant velocity and the sonic velocity are calculated by the relations:

$$G_{ct} = \frac{V}{\upsilon} \quad 23) \qquad\qquad c^2 = \left(\frac{\partial p}{\partial \rho}\right)_s \quad (24)$$

VALIDATION

The model was validated by comparing with published data. 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. The solution also yields quality, pressure, temperature and enthalpy distributions.

Very little experimental data have been published for diabatic case of a capillary tube-suction line heat exchanger. Figures 4 and 5 show good agreement between the mass flow rate calculated by the model and the data presented by Christensen and Jorgensen (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.

SIMULATION AND DESIGN RESULTS

Figures 6 and 7 illustrate the use of the capillary tube-suction line heat exchanger model for simulation. Figure 6 suggests that HFC-134a will respond similarly to CFC-12 as condensing temperature changes. Figure 7 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. The differences in thermodynamic and transport properties of CFC-12 and HFC-134a are reflected in this plot.

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 8 shows the capillary tube diameter calculation. for different condensing temperatures. The length of the heat exchanger is taken as the independent variable in Figure 9 to demonstrate its linear dependence on, and sensitivity to, the design condensing temperature. 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.

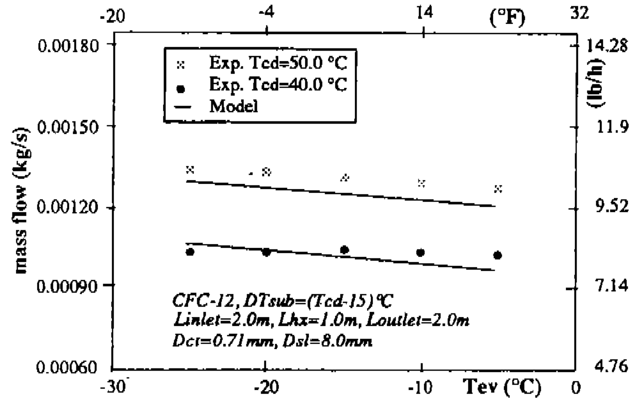

Figure 4. Validation of ct-slhx model (long entrance section)

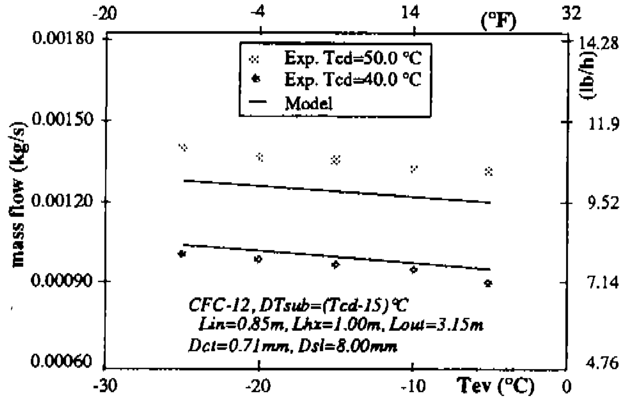

Figure 5. Validation of ct-slhx model (short entrance section)

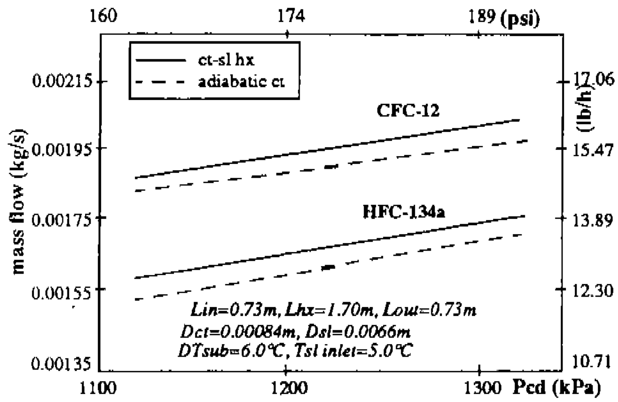

Figure 6 Effect of the condenser pressure on mass flow rate calculation (lateral ct-sl hx)

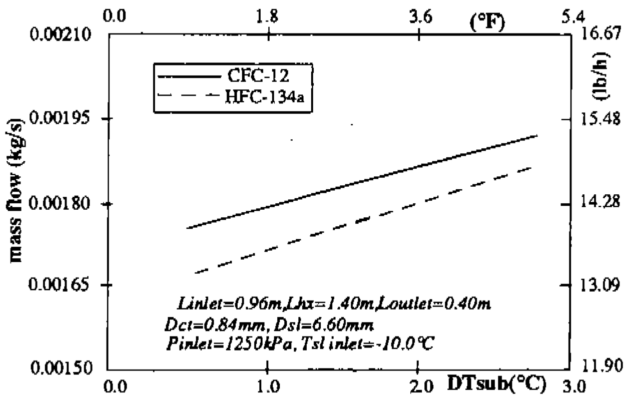

Figure 7 Effect of the inlet subcooling on mass flow simulation (lateral ct-sl hx)

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 10. The ASHRAE handbook cautions that heat exchanger region location can cause instabilities in refrigerator operation and this fact can be related to refrigerant recondensation. 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 as assumed by Pate and Tree (1984a) is unlikely to exist in most cases.

Finally Figure 11 illustrate difference 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.

CONCLUSIONS

The finite difference model presented here shows reasonably good agreement with published data for diabatic capillary tubes. 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. For a given mass flow rate the calculations suggest that HFC-134a will require a smaller capillary tube diameter and larger heat exchange length. Finally the model illustrates how the mass flow rate 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.

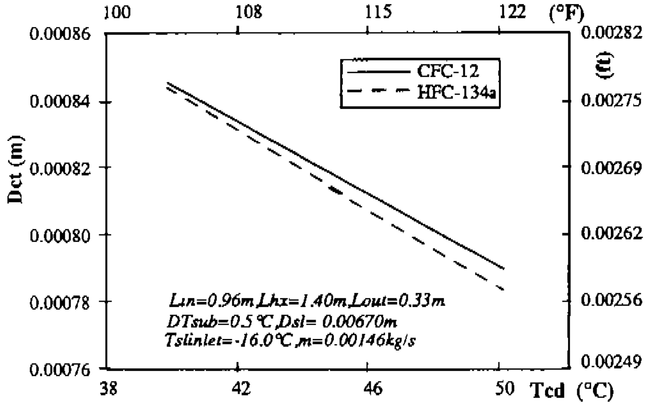

Figure 8 Effect of condenser temperature in the capillary tube diameter calculation (lateral ct-sl hx)

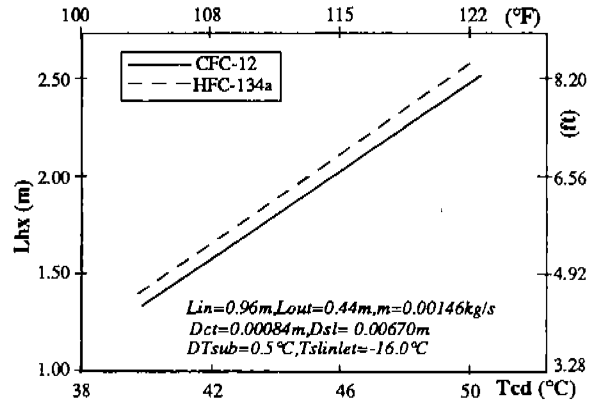

Figure 9 Effect of condenser temperature in the heat exchange length calculation (lateral ct-sl hx)

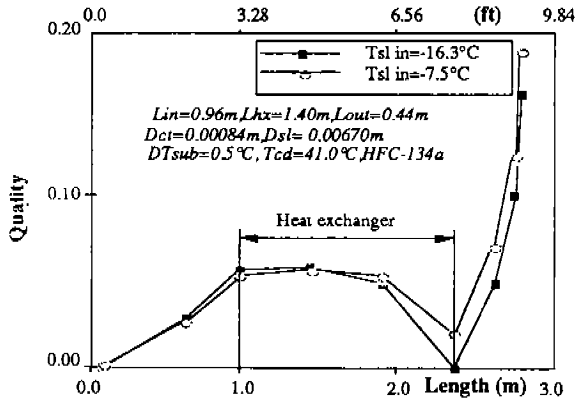

Figure 10 Quality profile in the capillary tube (lateral ct-sl hx)

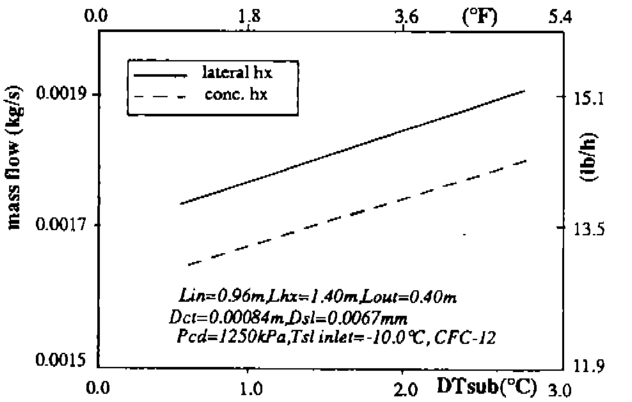

Figure 11 Mass flow calculation for lateral and concentric tubes ct-sl hx

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