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NUMERICAL AND EXPERIMENTAL STUDIES ON ADIABATIC AND NONADIABATIC CAPILLARY TUBES WITH HFC-134a

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ABSTRACT

This paper describes the numerical and experimental studies performed on capillary tubes with HFC-134a. The present study focuses on the effects of suction line heat exchange on flow characteristics of capillary tubes for simulating real operating conditions of household refrigerators. The capillary tube-suction line heat exchanger assembly considered in the present study is a double-pipe arrangement in which a portion of capillary tube passes inside the suction line to form a counter flow heat exchanger. In this assembly, superheated refrigerant vapor passes from the evaporator outlet to the compressor through the concentric circular-tube annulus. Experiments were conducted for both adiabatic and nonadiabatic capillary tubes to validate the numerical model. The flow rates predicted by the numerical model show good agreement with the present experimental data and with recent adiabatic measurements reported in literature.

NOMENCLATURE

A	Cross-sectional area
C_f	Friction factor
c_p	Constant pressure specific heat
D	Diameter
G	Mass flux
h	Refrigerant enthalpy
K	Boltzmann's constant
$\dot{m}$	Mass flow rate
Nu	Nusselt number
P	Pressure
Pr	Prantl number
q'	Heat transfer rate per unit length
Re	Reynolds number, $Re = \frac{GD}{\mu}$
T	Temperature
v	Specific volume
x	Vapor quality
z	Distance from capillary inlet
ε	Surface roughness height
μ	Viscosity
ρ	Density
σ	Surface tension

Subscripts

$crit$	Critical
f	Liquid phase
g	Vapor phase
s	Saturated
sc	Subcooled
sh	Superheated
$suct$	Suction line

INTRODUCTION

Household refrigerators generally operate with a capillary tube expansion device. In such applications, capillary tubes are often in thermal contact with suction line to form a counterflow heat exchanger. This configuration provides heat exchange between superheated refrigerant vapor passing through suction line to compressor and relatively warm refrigerant passing through capillary tube. Such heat exchange improves the thermodynamic efficiency of refrigeration by reducing refrigerant quality before evaporator inlet. The experimental and numerical studies conducted by Pate and Tree [1,2] using CFC-12 indicate that flow behavior in a capillary tube with suction line heat exchange differs considerably from that of an adiabatic capillary tube. The physical situation is such that two-phase flashing caused by friction effect is suppressed by the cooling effect of the gaseous refrigerant passing through suction line.

NUMERICAL MODEL

The following simplifying assumptions are made in formulating the description of flow through the capillary tube-suction line assembly: i) the capillary tube is a straight, horizontal, constant inner diameter tube, ii) the capillary tube is perfectly centered within the suction line, iii) negligible axial conduction in the fluid and tube walls, and iv) entrance effects are negligible.

Single Phase Flow Region

For adiabatic capillary tubes, location of flash point or liquid length is determined by the following equation [3]:

$$z_v = \frac{\rho_f D}{2C_f G^2}\left[(P_i - P_s) + (P_s - P_v)\right] \quad . \tag{1}$$

Here, the term $(P_i - P_v)$ is the pressure difference between the capillary tube inlet and the thermodynamic saturation pressure corresponding to the refrigerant inlet temperature, whereas $(P_s - P_v)$ is the pressure difference defined as the underpressure of vaporization. Chen et al. [3] established the following empirical correlation for the determination of this pressure difference :

$$\frac{(P_s - P_v)\sqrt{KT_s}}{\sigma^{3/2}} = 0.679\left[\frac{v_g}{v_g - v_f}\right] Re^{0.914}\left[\frac{T_{sc}}{T_{crit}}\right]^{-0.208}\left[\frac{D}{D'}\right]^{-3.18} \quad , \tag{2}$$

where D' is a reference length given by $D' = \sqrt{KT_s/\sigma} \times 10^4$. In the present study, liquid metastable length or delay of vaporization is determined by the cautious use of equation (2) for both adiabatic and nonadiabatic flow situations. However, for those situations in which equilibrium flash point takes place in the heat exchanger region of the capillary tube, metastable flow effects are not included in the computations. The simulation model does not account for the nonequilibrium two-phase flow occurring immediately downstream of the actual flash point.

In the present work, single-phase friction factor is calculated by well-known Colebrook's correlation:

$$\frac{1}{\sqrt{C_f}} = 2.28 + 4\log\left(D/\varepsilon\right) - 4\log\left[1 + \frac{4.65}{Re\left(\varepsilon/D\right)\sqrt{C_f}}\right] \quad . \tag{3}$$

Two-Phase Flow Region

The governing equations for steady one-dimensional homogeneous equilibrium two-phase flow through the capillary tube are obtained by applying basic conservation principles to differential control volumes. The subsequent numerical solution is facilitated by transforming the derived equations into the following form [4]:

$$-\frac{dP}{dz} = \frac{\dfrac{2C_f G^2}{\rho_m D} + \dfrac{G v_{fg}}{h_{fg} A} q'_{cap}\phi}{1 + \phi G^2\left[x\dfrac{\partial v_g}{\partial P} + (1-x)\dfrac{\partial v_f}{\partial P} + v_{fg}\left(\dfrac{\partial x}{\partial P}\right)_h\right]} \quad , \tag{4}$$

$$\frac{dh}{dz} = \phi\left\{\frac{q'_{cap}}{GA} - \frac{G^2}{\rho_m}\frac{dP}{dz}\left[x\frac{\partial v_g}{\partial P} + (1-x)\frac{\partial v_f}{\partial P} + v_{fg}\left(\frac{\partial x}{\partial P}\right)_h\right]\right\} , \tag{5}$$

where $\phi = \dfrac{1}{1 + G^2 v_{fg}/h_{fg}\rho_m}$.

In the present study, two-phase friction factor is calculated by the single-phase friction factor correlation given in equation (3) in which Reynolds number is based on the equivalent mixture viscosity μ_e which is defined according to McAdams:

$$\frac{1}{\mu_e} = \frac{x}{\mu_g} + \frac{(1-x)}{\mu_f} \quad . \tag{6}$$

Suction Line Heat Exchanger

The energy equation for steady one-dimensional refrigerant gas flow through the concentric circular-tube annulus is:

$$c_p \dot{m} \frac{dT_{suct}}{dz} = q'_{cap} + q'_{amb} \tag{7}$$

Here, q'_{cap} and q'_{amb} are the heat transfer rates per unit length from the capillary tube outer surface to the suction line gas and between the suction line gas and ambient through the suction line insulation, respectively. For the turbulent flow in the annulus system considered here, Nusselt number Nu for the outer surface of the capillary tube can be determined by the following relation [5]:

$$Nu / Nu_{tube} = 0.86 (D_{suct} / D_o)^{0.16} \tag{8}$$

Here, Nu_{tube} is the Nusselt number for the tube flow, D_o is the outer diameter of capillary tube, and D_{suct} is the inner diameter of suction line. The Nusselt numbers used in equation (8) are based on the hydraulic diameter D_h defined according to $D_h = D_{suct} - D_o$. The Nusselt number for the fully developed tube flow is determined using the correlation given by Gnielinski [5]:

$$Nu_{tube} = \frac{(Re - 1000) Pr C_f}{1.0 + 12.7 \left(Pr^{2/3} - 1 \right) \sqrt{C_f / 2}} \tag{9}$$

where C_f is the friction factor for fully developed turbulent flow in a smooth pipe, $C_f = (1.58 \ln Re - 3.28)^{-2}$.

Numerical Solution

Two-phase flow model described by equations (4), (5) and (7) is an initial value problem. Fourth-order Runge-Kutta algorithm is used to advance the solution one step forward in z direction, with the known conditions at the previous z location. The procedure was repeated until either critical flow or evaporator pressure is reached. The critical flow condition is determined by using the whole denominator of equation (4). For a given adiabatic capillary tube, mass flow rate of refrigerant is determined by an iterative procedure. Mass flow rate is continuously adjusted until the calculated tube length equals to the prescribed tube length. For the capillary tube-suction line assembly considered here, additional iteration loop is required on suction line exit temperature to match the calculated suction line inlet temperature to the prescribed conditions at the evaporator outlet.

EXPERIMENTAL SETUP AND PROCEDURE

A schematic diagram of the experimental setup is shown in Figure 1. The test system is a basic refrigeration circuit with provisions to control the operating conditions of capillary tube-suction line test section. The test unit is equipped with water cooled condenser and subcoolers to adjust the pressure and temperature of liquid refrigerant entering the capillary tube. Evaporator is secondary refrigerant calorimetric vessel which is defined in ISO-917. For nonadiabatic tests, superheating value of refrigerant vapor entering the suction line is controlled by varying the heat input to secondary fluid. The entire length of the test section was insulated to minimize the heat transfer to ambient.

The parameters measured in the tests were inlet and outlet conditions of the test section and refrigerant flow rate. Temperature measurements were conducted using RTD probes (accuracy: ±0.4°F) and type T thermocouples (accuracy: ±0.4°F). RTD probes were inserted directly into the refrigerant line. Precise pressure gauges were used for the pressure measurements at the inlet and outlet of the capillary tube (ranges: 0-500 PSIA, 0-150 PSIA accuracy: 0.1% FS). Refrigerant flow rate was measured using a coriolis type mass-flow meter (range:0-2 lb/min accuracy: 0.2 % FS). Data were recorded using a data acquisition system at one minute time intervals for a period of at least one hour after steady working conditions were reached.

In the present work, total of 146 tests were run using copper tubes with 0.66 and 0.8 mm diameters. The outer diameter of the test tubes was 2.0 mm. The tested tubes were standard refrigeration capillary tubes taken from the same manufacturer. Of all the tests run in this study, 83 were for adiabatic tubes. The range of geometric parameters selected in this study was motivated by design applications. For the adiabatic flow case, 3500, 4500 and 5500 mm long tubes were tested. The nonadiabatic tests were run using the capillary tubes of 5500 and 6500 mm. The suction line heat exchanger was 1700 mm long and located at 400 mm from the capillary tube exit for each nonadiabatic test run. The inner diameter of suction line tube was 5.6 mm. The refrigerant pressure at the capillary tube inlet was adjusted to the saturation values corresponding to the condenser temperatures of 43.3, 48.9, 54.4 °C (110, 120, 130 °F). Liquid subcooling at capillary tube inlet were varied from 5 to 20 °C approximately for each condenser temperature.

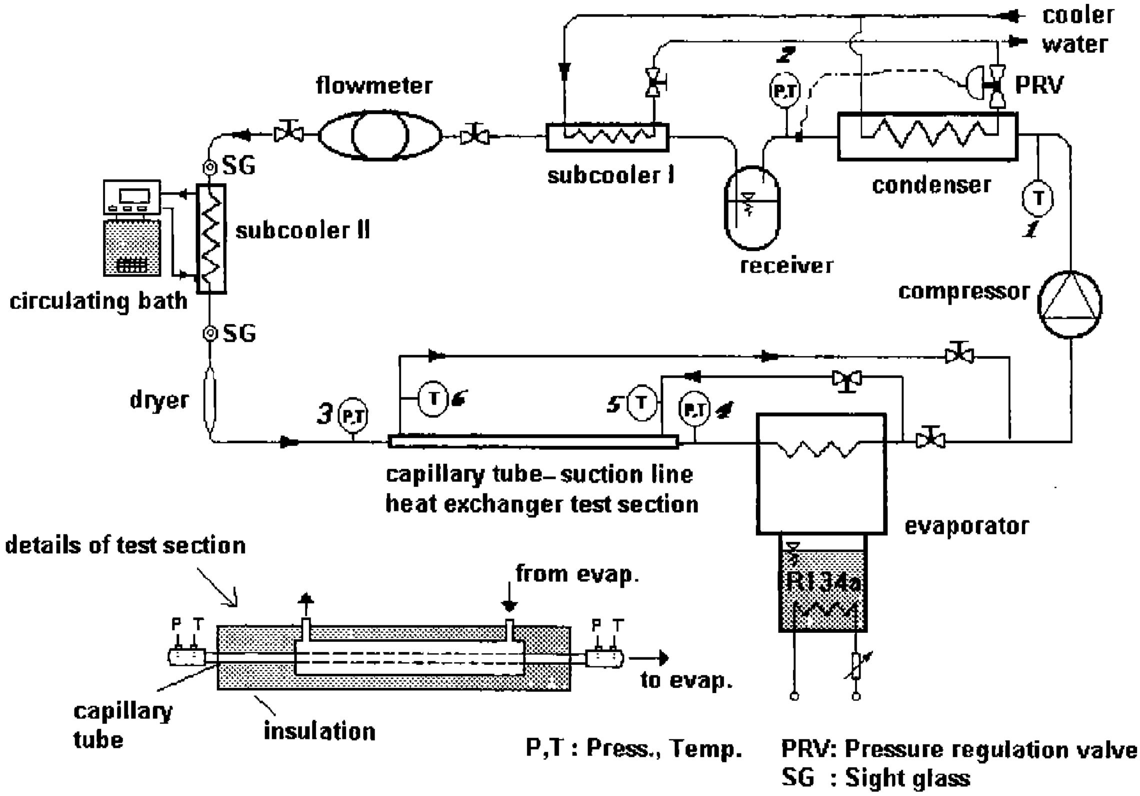

Figure 1. Schematic diagram of the experimental setup.

RESULTS AND DISCUSSIONS

The calculated results were compared with the measured data to validate the numerical model. In the present study, the roughness height of capillary tubing is taken as ε =0.00046 mm (0.018 inches) which was also used in [6]. Thermodynamic and transport properties of HFC-134a are determined by the equations and tables of [7]. Figures 2(a) and 2(b) show comparisons of the predicted and measured refrigerant flow rates for adiabatic and nonadiabatic test runs, respectively. As seen from these figures, the flow rates predicted by the numerical model are within 10 percent of the measurements for most of the test cases. Table 1 summaries the test data obtained in tubes of 0.66 and 0.8 mm diameters and 5500 mm long. Also included in the table are the corresponding predictions by the numerical model.

Comparisons were also made with recent experimental data reported by Wijaya [8,9]. Wijaya ran a number of adiabatic tests with HFC-134a over a similar range of operating conditions. Tested tubes were 0.026, 0.031 and 0.033 inches in diameter and from 5 to 10 feet in length. Wijaya presented his experimental data in the form of simple correlations [9]. The predictions of the numerical model that is presented in this paper also showed similarly good agreement with Wijaya's adiabatic test data [4].

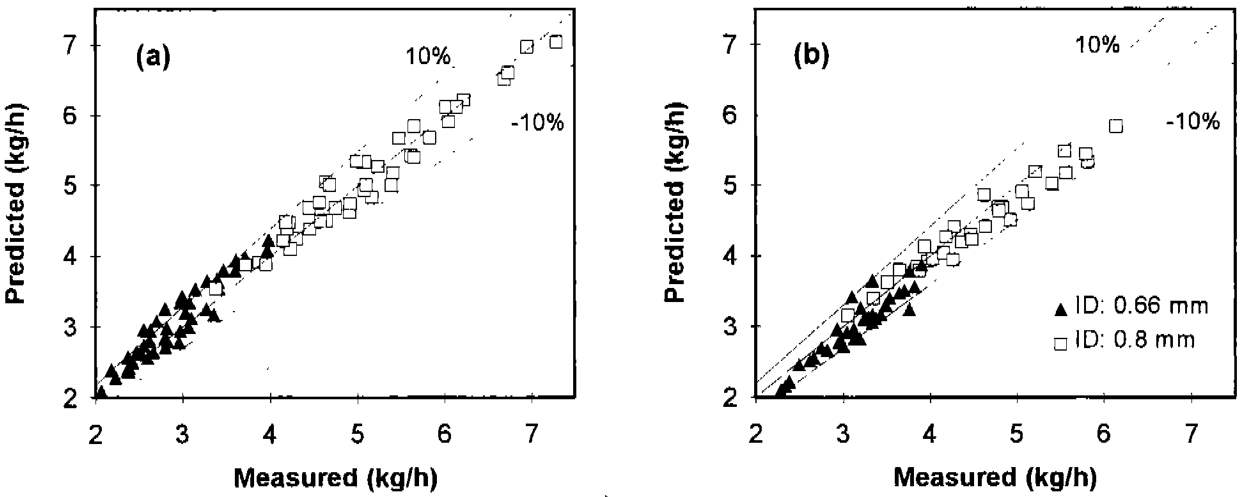

Figure 2. Comparison of the predicted and measured refrigerant flow rates.
(a) Adiabatic. (b) Nonadiabatic.

Table 1. Typical test data.
(a) Tube ID: 0.66 mm, length: 5500 mm.

adiabatic						nonadiabatic					
T_{con} (°C)	T_{sc} (°C)	T_{evap} (°C)	m_{exp} (kg/h)	m_{num} (kg/h)	Diff. %	T_{sc} (°C)	T_{evap} (°C)	T_{sh} (°C)	m_{exp} (kg/h)	m_{num} (kg/h)	Diff. %
43.3	5.2	-29.9	2.07	2.07	0.0	3.7	-28.4	23.0	2.29	2.10	-8.3
	8.7	-29.0	2.24	2.25	0.4	4.7	-28.2	22.2	2.34	2.16	-7.7
48.9	6.6	-28.0	2.38	2.34	-1.7	5.7	-26.5	21.4	2.49	2.46	-1.2
	8.8	-27.2	2.43	2.46	1.2	6.9	-25.9	15.8	2.66	2.56	-3.8
54.4	7.4	-26.0	2.53	2.59	2.4	5.7	-24.0	15.1	2.75	2.70	-1.8
	14.7	-23.5	3.07	2.95	-3.9	9.2	-22.7	4.7	3.12	2.97	-4.8

(b) Tube ID: 0.8 mm, length: 5500 mm.

adiabatic						nonadiabatic					
T_{con} (°C)	T_{sc} (°C)	T_{evap} (°C)	m_{exp} (kg/h)	m_{num} (kg/h)	Diff. %	T_{sc} (°C)	T_{evap} (°C)	T_{sh} (°C)	m_{exp} (kg/h)	m_{num} (kg/h)	Diff. %
43.3	5.7	-19.7	3.38	3.49	3.3	8.2	-18.4	5.0	4.37	4.19	-4.1
	9.6	-17.2	3.95	3.85	-2.5	9.1	-18.1	5.0	4.47	4.30	-3.8
48.9	5.7	-17.7	3.71	3.80	2.4	6.7	-16.7	1.0	4.28	4.41	3.0
	9.6	-15.9	4.15	4.18	0.7	9.1	-14.5	0.5	4.84	4.68	-3.3
54.4	8.4	-15.0	4.22	4.40	4.3	7.3	-17.0	0.4	4.63	4.86	5.0
	10.8	-14.2	4.44	4.44	3.7	10.2	-14.7	0.7	5.21	5.19	-0.4

Calculated pressure and vapor quality profiles for one representative case are presented in Figures 3(a) and 3(b). The capillary tube considered in this case is 0.66 mm in diameter and 5500 mm long. The condenser and evaporator temperatures are 54.4°C and -24.0°C, respectively. Liquid subcooling at capillary tube inlet is 5.7°C. Refrigerant vapor at -8.9°C enters the suction line heat exchanger. Measured and predicted flow rates for this case are 2.75 and 2.70 kg/h, respectively. The equilibrium flash point occurs at 1810 mm from the capillary tube inlet. Temperatures measured at various positions upstream of suction line heat exchanger are shown in Figure 3(a). Temperatures were measured by thermocouples soldered to the tube wall. As seen from this figure, there is considerable delay in flashing. The predicted and measured values for the metastable liquid length are 471 mm and roughly 1000 mm, respectively. For this case, the delay of vaporization is underpredicted by equation (2). Also shown in Figure 3(a) is the calculated suction line temperature profile. The increase in the suction line gas temperature through the heat exchanger length is 40.6°C. The flow rate predicted for the corresponding adiabatic operation of the tube is 2.5 kg/h; this value is 7.4% less than the nonadiabatic operation.

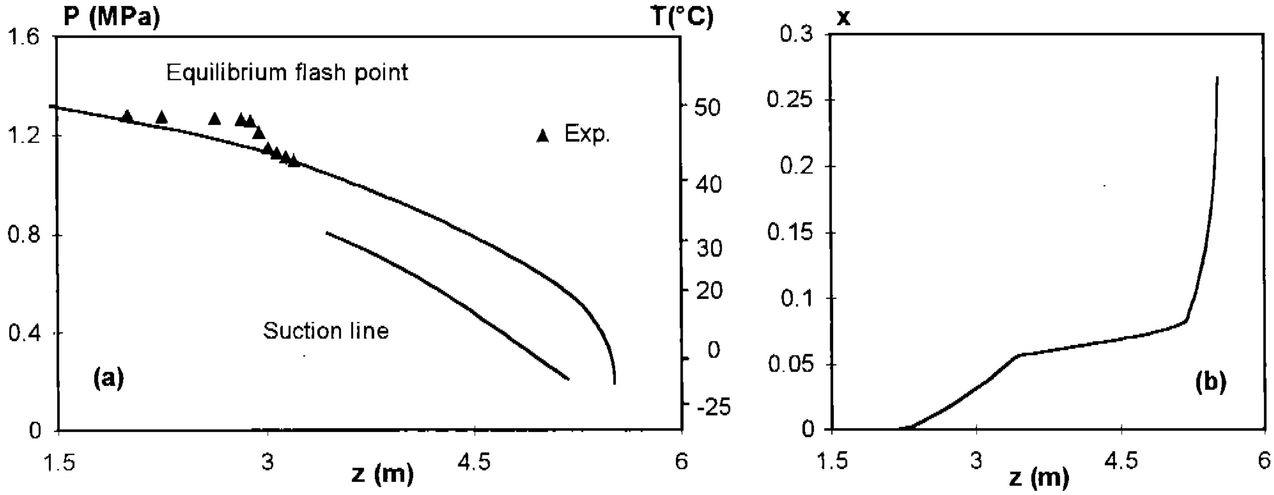

Figure 3. Typical profiles for nonadiabatic operation, 0.66 mm diameter by 5500 m long tube.
(a) Pressure and temperature. b) Vapor quality.

CONCLUSIONS

The present study was intended to develop a general numerical model to simulate the effects of suction line heat exchanger on performance characteristics of capillary tubes. Experimental results were used to validate the numerical model for both adiabatic and nonadiabatic flow situations. The flow rates predicted by the numerical model show good agreement with the measurements over a wide range of operating conditions and geometric parameters. The established numerical model can be implemented in refrigeration cycle simulation programs without adding excessive CPU requirements.

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