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# EXPERIMENTAL AND NUMERICAL STUDIES ON ADIABATIC FLOW OF HFC MIXTURES IN CAPILLARY TUBES

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## ABSTRACT

This work presents the experimental and numerical studies on the adiabatic flow of HFC mixtures in capillary tubes. Two capillary tubes of inner diameter of 1.2 and 1.6mm with length of 1.5m were tested. The condensation temperatures of 40, 45, and 50°C, and the subcooling temperatures ranging from 3 to 12°C were chosen as operating conditions. The numerical model developed in this study has considered the effects of the underpressure for vaporization, kinetic energy of the refrigerant during expansion, and roughness of capillary tube. This work presents characteristic charts and correction factors for HFC-32/125/134a.

## NOMENCLATURE

| | | | | | |
|---|---|---|---|---|---|
| $c_1$~$c_4$ | Coefficient in equation (2) | Re | Reynolds number ($=Gd/\mu$) | Subscripts | |
| $D$ | Inner diameter of condenser, m | $T$ | Temperature, °C | bubble | Bubble point |
| $d$ | Inner diameter of capillary tube, m | $\Delta T_{sub}$ | Degree of subcooling, °C | cond | Condensation |
| $f$ | Friction factor | $v$ | Specific volume, $m^3/kg$ | dew | Dew point |
| $G$ | Mass flux, $kg/m^2s$ | $z$ | Axial coordinate, m | $f$, $g$ | Saturated liquid, vapor |
| $h$ | Specific enthalpy, kJ/kg | $\varepsilon$ | Roughness, m | in | Inlet |
| $L_L$ | Length of liquid region, m | $\mu$ | Viscosity, Pa s | sat | Saturation |
| $P$ | Pressure, kPa | $\rho$ | Density, $kg/m^3$ | TP | Two-phase |
| $q$ | Quality | | | vap | Vaporization |

## INTRODUCTION

Capillary tube is a common expansion device in refrigeration system. It is a simple tube, but strongly affects the performance of the system. Therefore it is necessary to choose a proper capillary tube for an optimal performance. Researches about capillary tubes have been done, and rating curves [1] to select a capillary tube are available. However, most deal with concern CFC-12 and HCFC-22, which will be phased out due to the ozone depletion, and the rating curves [1] do not consider the effect of inside wall roughness of capillary tube. Capillary tube is too slender to ignore the effect of roughness [2]. Therefore, studies about flow of alternative refrigerants in capillary tube and effects of roughness of the tube are definitely required. Most studies of alternative refrigerants are about HFC-134a [3, 4, 5], which is an alternative to CFC-12, but there are few studies about alternatives to HCFC-22.

In this study, an experimental and numerical analysis on adiabatic flow in capillary tubes has been done. We used HFC-32/134a (30/70, by mass), HFC-32/125 (60/40) and HFC-32/125/134a (23/25/52) as working fluids. To model the flow in capillary tube, we assumed two-phase flow as homogeneous flow, and considered the effects of a metastability and roughness of inside wall. The correction factor curves for geometry and roughness of capillary tube as well as basic rating curves were presented.

## EXPERIMENTAL APPARATUS AND PROCEDURE

Experimental apparatus is composed of condenser, compressor, evaporator and capillary tube as an expansion device. Those are general components of refrigerating system. The schematic diagram is shown in Fig. 1. The condensation and evaporation pressures can be varied by adjusting the temperature and mass flow rate of water that is used as a secondary heat transfer fluid. The degree of subcooling is changed by using the subcooler and PID controlled electric heater. The mass flow rate of refrigerants is measured by mass flow meter (accuracy: 0.48%), pressure and temperature are measured by

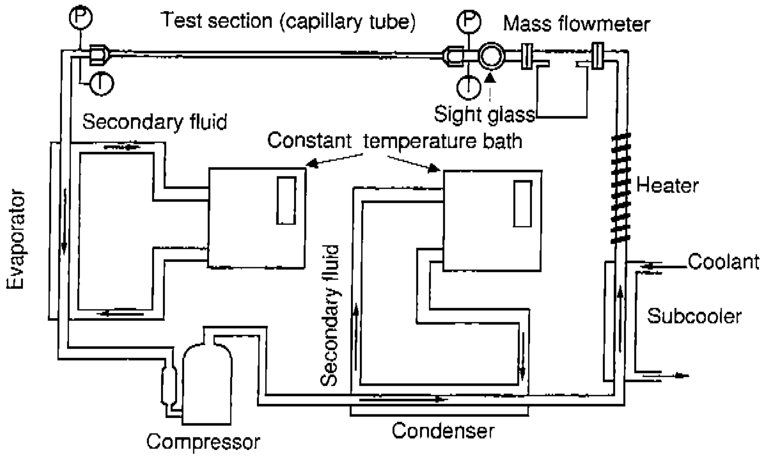

Fig. 1 Schematic diagram of experimental apparatus.

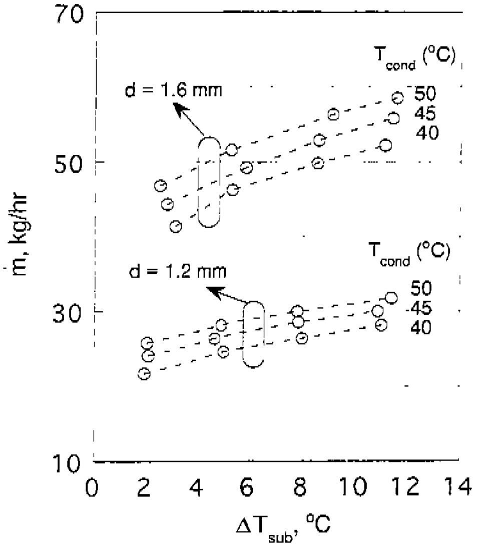

Fig. 2 Variation of mass flow rate with respect to the degree of subcooling.

pressure transducer (accuracy: 0.2%) and T-typed thermocouple, respectively. Table 1 shows the specification of capillary tubes and experimental conditions.

When the thermocouple is inserted in the tube, it will affect the flow because of the slendemess of the capillary tube. Therefore, the outside wall temperature is measured at 21 points and used to calculate the temperature of refrigerant in the capillary tube. For pressure measurement, 11 holes on the capillary tube are made by laser beams to avoid any chips remaining inside the capillary tube. The diameter of these holes is about 0.1 mm. These holes are covered with copper tube whose inner diameter is larger than that of capillary tube and a pressure tap was made on the copper tube. Reference [6] describes details of pressure measurements.

In order to check the validity of choosing the wall temperature as the refrigerant temperature, the measured pressure is compared with the saturation pressure calculated from the corresponding wall temperature. The measured pressure is well fit for the saturated pressure within 1.5 % error bound [6] at the maximum. It is available to choose the wall temperature as the temperature of refrigerant in the capillary tube. In this study, the saturated pressure and other thermodynamic properties of refrigerants are calculated by modified Carnahan-Starling equation of state [7].

## EXPERIMENTAL RESULTS

The mass flow rate in capillary tube is mainly affected by condensation temperature, the degree of subcooling, and the geometry of capillary tube such as length and inner diameter. The outlet condition of capillary tube does not affect the mass flow rate because the flow at outlet is choked in this study. Condensation temperature is determined as an average of

Table 1 Experimental conditions in this study

| Capillary tube | | I | II |
|---|---|---|---|
| | Inner diameter (mm) | 1.2 | 1.6 |
| | Length (m) | 1.5 | 1.5 |
| | Roughness (μm) | 0.1963 | 0.6894 |
| Inlet pressure (kPa) | 1540 ~ 2800 | | |
| Condensation temperature (°C) | 40, 45, 50 | | |
| Degree of subcooling (°C) | 2 ~ 12 | | |
| Mass flux (kg/m$^2$s) | 3980 ~ 10060 | | |

dew point and bubble point temperature as shown in equation (1).

$$T_{\text{cond}} = \frac{T_{\text{dew}} + T_{\text{bubble}}}{2} \tag{1}$$

Figure 2 shows the effect of condensation temperature and subcooling on the mass flow rate of HFC-32/125/134a. Mass flow rate in capillary tube increases as the degree of subcooling increases when condensation temperature and the inner diameter of capillary remain constant. The mass flow rate increases as condensation temperature becomes higher and the inner diameter becomes bigger when the degree of subcooling remains constant. The rise of condensation temperature causes the pressure at the inlet of capillary tube to be higher, and the mass flow rate increases.

Figure 3 shows the comparison of mass flow rate with that of HCFC-22 in capillary tube with 1.6 mm inner diameter when condensation temperature is 45 °C. The mass flow rate of HFC-32/125/134a and HFC-32/134a is similar to that of HCFC-22, but that of HFC-32/125 is larger than that of HCFC-22. HFC-32/125/134a and HFC-32/134a are zeotropic mixtures whose vapor pressures are similar to that of HFC-22 while HFC-32/125 is an azeotropic mixture whose vapor pressure is higher than that of HCFC-22. The high pressure at the inlet of capillary tube makes the mass flow rate of HFC-32/125 be larger than that of HCFC-22.

## Underpressure for Vaporization

Theoretically flashing point in a capillary tube is the point where pressure becomes the vapor pressure corresponding to inlet temperature. However, nucleation does not start at the flashing point, vaporization is actually delayed [8]. Ignorance of the delay of vaporization makes errors when the flow in capillary tube is analyzed. Figure 4 shows the temperature and pressure variation along the capillary tube for HFC-32/125/134a. Solid circles represent saturation pressures corresponding to temperatures. Region I is a stable liquid state, region II is a metastable liquid state in Fig. 4. Underpressure for vaporization is a difference between the vapor pressure corresponding to inlet temperature and the pressure where flashing actually starts. It is known that the underpressure for vaporization increases as the mass flux of refrigerants increases and the degree of subcooling decreases [8]. In this study, the underpressure is correlated as a function of the mass flux and subcooling as shown in equation (2). The coefficients of equation, $c_1$, $c_2$, $c_3$, $c_4$ are calculated from experimental data by least squares method and presented in Table 2. The validity of equation (2) is limited to the displayed range in Table 1 of mass flux and the degree of subcooling.

$$P_{\text{sat}} - P_{\text{vap}} = c_1 + c_2 \times G + c_3 \times \Delta T_{\text{sub}} + c_4 \times G \times \Delta T_{\text{sub}}. \tag{2}$$

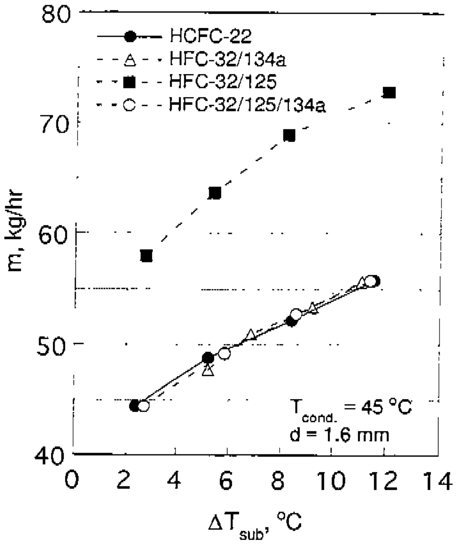

Fig. 3 Comparison of mass flow rates of refrigerants.

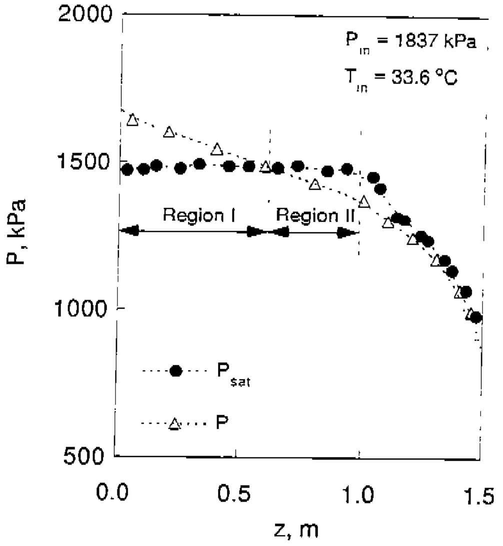

Fig. 4 The measured pressure and saturation pressure along the capillary tube for HFC-32/125/134a.

Table 2 Coefficients of correlation for underpressure

| Refrigerants | $c_1 \times 10^{-2}$ | $c_2 \times 10^{2}$ | $c_3 \times 10^{-1}$ | $c_4 \times 10^{5}$ |
|---|---|---|---|---|
| HFC-32/134a | -3.911112 | 7.427592 | 4.705004 | -8.561734 |
| HFC-32/125 | -3.173929 | 4.692433 | 4.920039 | -5.463006 |
| HFC-32/125/134a | -1.536972 | 3.844115 | 1.818622 | -3.036465 |

## NUMERICAL MODEL

In this study, the effects of the underpressure for vaporization and roughness of capillary tube are considered in numerical model. Kinetic energy term is included in energy equation of refrigerant flow in capillary tube because the kinetic energy might be too large to ignore in case that flow at outlet of capillary tube is choked [9]. The flow in capillary tube is divided into subcooled liquid region and vapor-liquid two phase region. The assumptions used in this model are listed as follows.

1) The cross sectional area of capillary tube is constant, and the tube is straight and horizontal.
2) Flow in capillary tube is one dimensional and adiabatic.
3) Liquid flow is incompressible.
4) Two phase flow is a homogeneous flow.
5) The refrigerant flow at outlet of capillary tube is choked.

### Single Phase Flow Region

From momentum equation, the length of subcooled liquid region or location of flash point is determined by the following equation,

$$L_L = \frac{\left(P_{\mathrm{cond}} - P_{\mathrm{in}}\right) + \left(P_{\mathrm{in}} - P_{\mathrm{sat}}\right) + \left(P_{\mathrm{sat}} - P_{\mathrm{vap}}\right)}{f\left(G^2/2\rho_f d\right)}, \tag{3}$$

$$\left(P_{\mathrm{cond}} - P_{\mathrm{in}}\right) = 0.42\left[1 - \left(d/D\right)^2\right]\frac{G^2}{2\rho_f}, \tag{4}$$

where $(P_{\mathrm{cond}} - P_{\mathrm{in}})$ is the pressure loss due to sudden contraction at inlet of capillary tube, and $(P_{\mathrm{sat}} - P_{\mathrm{vap}})$ is the underpressure for vaporization that is calculated by equation (2).

In this model, single phase friction factor is calculated by Haaland's equation [10]:

$$f = \left\{-1.8\log\left[\frac{6.9}{\mathrm{Re}} + \left(\frac{\varepsilon}{3.7d}\right)^{1.11}\right]\right\}^{-2}. \tag{5}$$

### Two Phase Flow Region

From momentum equation and energy equation concluding the effect of kinetic energy, the pressure drop of steady one-dimensional two phase homogeneous flow in capillary tube may be expressed as following equation,

$$\frac{\mathrm{d}P}{\mathrm{d}z} = \frac{-f\dfrac{G^2 v_{\mathrm{TP}}}{2d}}{\left[1 + G^2 q\dfrac{\mathrm{d}v_g}{\mathrm{d}P} - \dfrac{\left(\dfrac{\mathrm{d}h_f}{\mathrm{d}P} + q\dfrac{\mathrm{d}h_{fg}}{\mathrm{d}P} + qG^2 v_{TP}\dfrac{\mathrm{d}v_g}{\mathrm{d}P}\right)G^2}{\left(\dfrac{h_{fg}}{v_{fg}} + G^2 v_{\mathrm{TP}}\right)}\right]}. \tag{6}$$

In this model, two phase friction factor is also calculated by equation (5) in which Reynolds number is based on the two phase viscosity that is defined by Cicchitti *et al.* [11] such as,

$$\mu_{TP} = q\mu_g + (1-q)\mu_f . \tag{7}$$

## SIMULATION RESULTS AND DISCUSSIONS

The program for calculating the mass flow rate and pressure variation in capillary tube is developed when the condition of inlet and geometry of capillary tube are known. In the program, equation (3) is applied to the single phase liquid region and equation (6) is solved by 4th order Runge-Kutta method in the two phase region. The mass flow rate is assumed for the first computation, and reassumed again until the flow in capillary tube becomes choked. If the flow is choked, the computation is terminated.

### Comparison with Experimental Data

To validate the numerical model, the calculated results are compared with the existing experimental data. Figure 5 shows comparisons of the predicted and measured refrigerant flow rates of Whitesel, Kuehl & Goldschmidt [9], Melo *et al.* [4], and Dirik *et al.* [5]. The flow rates predicted by the numerical model are within 10 % of most of experimental data. Figure 4 shows the temperature and pressure variations of HFC-32/125/134a in capillary tube. Solid line represents calculated values and symbol means the experimental data. The calculated values accord with experimental data well.

### Characteristic Charts

The components of refrigeration system using alternative refrigerants have to be modified to get the same performance as system using HCFC-22, and the characteristic charts of alternative refrigerants like ASHRAE charts [1] are requireded to select the proper capillary tube. The characteristic charts of flows in capillary tube for HFC-32/125/134a (23/25/52) are made by the numerical model of this study. Figure 6 presents basic rating curve for smooth tube with inner diameter of 1.6 mm and length of 2 m. In the basic rating curves, the pressure at outlet of capillary tube is critical pressure at which flow is choked. Flow rate increases as subcooling increases while inlet pressure is constant and flow rate increases as inlet pressure increases while subcooling is constant.

Figure 7 presents capillary flow factor for geometry that means the ratio of flow rate of general tube to that of basic tube. The effect of diameter and length of tube may be calculated by Fig. 7. If length of tube becomes 5 m and diameter 1.5 mm, mass flow rate reduced by 50 % while other flow condition is not changed. The flow rate in capillary tube is affected by the roughness of inner wall as well as length and diameter of tube. Figure 8 presents capillary flow factor for relative roughness. The flow rate decreases by 20 % as relative roughness increases up to 0.007. The effect of roughness

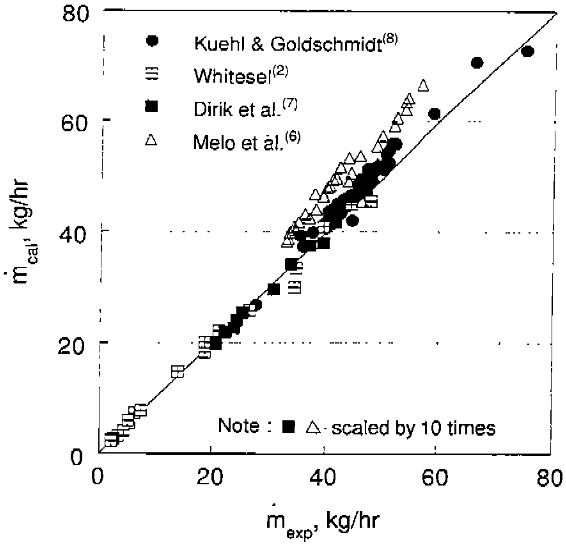

Fig. 5 Comparison of calculated mass flow rate with experimental data.

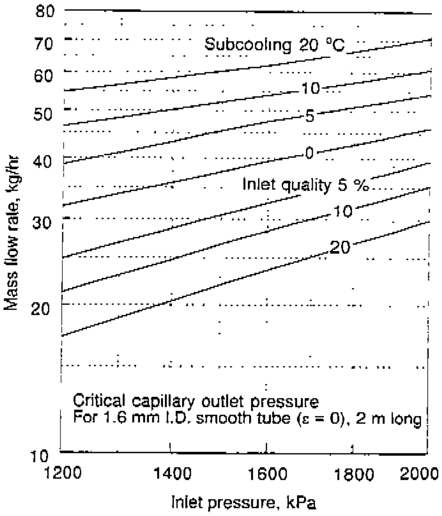

Fig. 6 Basic rating curves for capillary tubes (HFC-32/125/134a).

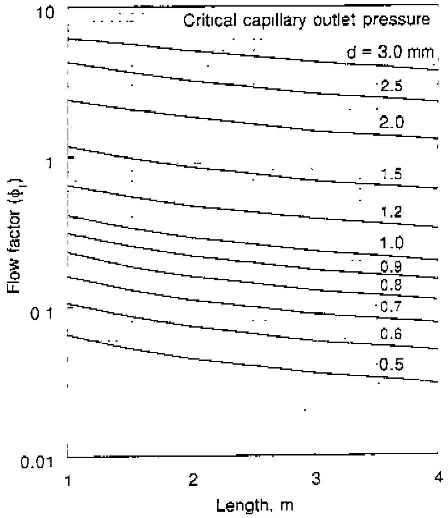

Fig. 7 Capillary flow factor for geometry of capillary tube (HFC- 32/125/134a).

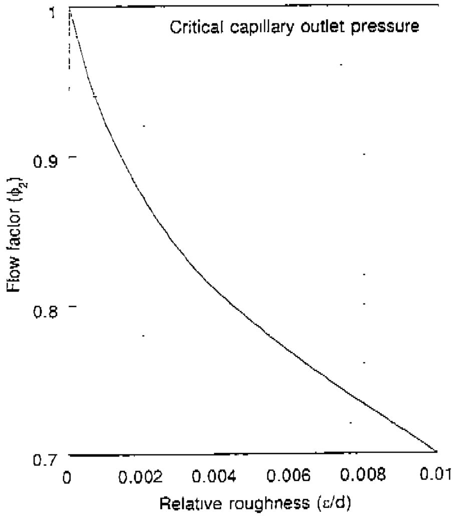

Fig. 8 Capillary flow factor for relative roughness of inside wall of capillary tube (HFC-32/125/134a).

## CONCLUSIONS

Experimental and numerical studies on adiabatic flow of HFC refrigerant mixtures in capillary tubes are performed. Metastable state in refrigerant flow through capillary tubes is experimentally investigated. Correlated equation for calculating underpressre is suggested. The flow rates predicted by the numerical model show good agreement with the experimental data of other researchers. Rating curves in this study can be a useful tool to select a proper capillary tube for refrigeration system using HFC mixture as working fluid.

## REFERENCES

[1] ASHRAE, 1994, ASHRAE Handbook Refrigeration.

[2] Lin, S., Kwok, C. C. K., Li, R.-Y., Chen, Z.-H. & Chen, Z.-Y., 1991, "Local frictional pressure drop during vaporization of R-12 through capillary tubes," Int. J. Multiphase Flow, Vol. 17, pp. 95~102.

[3] Wijaya, H., 1992, "Adiabatic capillary tube test data for HFC-134a," Proc. of Int. Refrig. Conf., Purdue University, U. S. A., Vol. 1, pp. 63~71.

[4] Melo, C., Ferreira, R. T. S., Neto, C. B., Goncalves, J. M., and Thiessen, M. R., 1994, "Experimental analysis of capillary tubes for CFC-12 and HFC-134a," Proc. of Int. Refrig. Conf., Purdue University, U. S. A., pp. 347~352.

[5] Dirik, E., Inan, C., and Tanes, M. Y., 1994, "Numerical and experimental studies on adiabatic and nonadiabatic capillary tubes," Proc. of Int. Refrig. Conf., Purdue University, U. S. A., pp. 365~370.

[6] Chang, S.-D. and Ro, S. T., 1996, "Pressure drop of pure HFC refrigerants and their mixtures flowing in capillary tubes," Int. J. Multiphase Flow (accepted).

[7] Kim, M. S., Kim, T. S., and Ro, S. T., 1991, "Estimation of thermodynamic properties of refrigerant mixtures using a modified Carnahan-Starling equation of state," Trans. Kor. Soc. of Mech. Eng., Vol 15, pp. 2189~2205 (in Korean).

[8] Li, R. Y., Lin, S., Chen, Z. Y., and Chen, Z. H., 1990, "Metastable flow of R12 through capillary tubes," Int. J. Refrig., Vol. 13, pp. 181~186.

[9] Kuehl, S. J. and Goldschmidt, V. W., 1991, "Modeling of steady flows of R-22 through capillary tubes," ASHRAE Trans., Vol. 97, Part 1, pp. 139~148.

[10] Haaland, S. E., 1983, "Simple and explicit formulas for the friction factor in turbulent pipe low," J. Fluids Eng., Vol. 105, pp. 89~90.

[11] Cicchittii, A., Lombardi, C., Silvestril, M., Soldaini, G, and Zavalluilli, R., 1960, "Two-phase cooling experiments-pressure drop, heat transfer and burnout measurements," Energia Nucleare, Vol. 7, pp. 407~425.

[12] Whitesel, H. A., 1957, "Capillary two-phase flow, Part II," Refrigeration Engineering, Vol. 65, pp. 35~40.