



PRESSURE DROP OF PURE HFC REFRIGERANTS AND THEIR MIXTURES FLOWING IN CAPILLARY TUBES

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Abstract—The pressure drop of a capillary tube flow is experimentally investigated. In this study, pure refrigerants such as R32, R125 and R134a and their mixtures such as R32/R134a (30/70 by mass fraction), R32/R125 (60/40), R125/R134a (30/70) and R32/R125/R134a (23/25/52) are used as test fluids. The binary interaction parameters for viscosities of the liquid state of refrigerant mixtures are found based upon the experimental data in the open literature. Several homogeneous flow models predicting the viscosity of two-phase region are compared and Cicchitti's equation is shown to be the most adequate for adiabatic capillary flows. A model for the prediction of the frictional pressure drop of single and two-phase flow is presented for the refrigerant mixtures in this study. Copyright © 1996 Elsevier Science Ltd.

Key Words: friction factor, capillary tube, HFC refrigerant, refrigerant mixture

1. INTRODUCTION

After CFC (chlorofluorocarbon) and HCFC (hydrochlorofluorocarbon) refrigerants were known to provide a principal cause to ozone depletion and global warming, production and use of these refrigerants have become restricted (UNEP 1987), even though these refrigerants have excellent chemical and thermodynamic properties. As an alternative to R22, which is one of HCFC refrigerants and is used for an air-conditioner or a heat pump, refrigerant mixtures are now being suggested for drop-in and eventual replacement (ARI 1992). These mixtures are composed of HFC (hydrofluorocarbon) refrigerants having zero ODP (ozone depletion potential). The refrigeration system using new alternative refrigerants have to be modified because transport properties of these alternative refrigerants differ from those of conventional refrigerants even though thermodynamic properties are similar to each other.

Capillary tube which is used as an expansion device of the vapor-compression refrigeration system has to be modified, too. It has advantages of simplicity, inexpensiveness, and durability caused by having no moving parts. However, since the capillary tube is not adjustable to changing load conditions, there is the disadvantage that an efficiency decreases when load conditions change. In redesigning the system using alternative refrigerants, therefore, it is very important to choose the capillary tube which matches the other consisting elements. It is necessary to predict the pressure drop of alternative refrigerants flowing in the capillary tube because one of the main functions of the capillary tube is lowering the pressure to achieve the low temperature. Two causes of the pressure drop in the capillary tube are friction on the wall and acceleration due to the variation of specific volume. The friction is known to be the principal cause and thus the friction factor of refrigerant flow is very important in designing a capillary tube.

The flow in the capillary tube can be divided into a subcooled liquid region before vaporization and a two-phase region after the refrigerant starts vaporization. It is reasonable to analyse the two-phase flow through the capillary tube by using the homogeneous flow model since the two-phase flow pattern in the capillary tube is a bubbly flow (Koizumi & Yokoyama 1980) of exit quality 0.2–0.3. However, the quality of the refrigerant does not vary linearly along the tube, so the two-phase flow in capillary tube was analysed by local friction factor.

In this study, pressure drop of HFC refrigerants and their mixtures flowing in capillary tubes was experimentally investigated to provide data for designing a capillary tube suitable for refrigeration system using alternative refrigerants. The friction factor that is an important

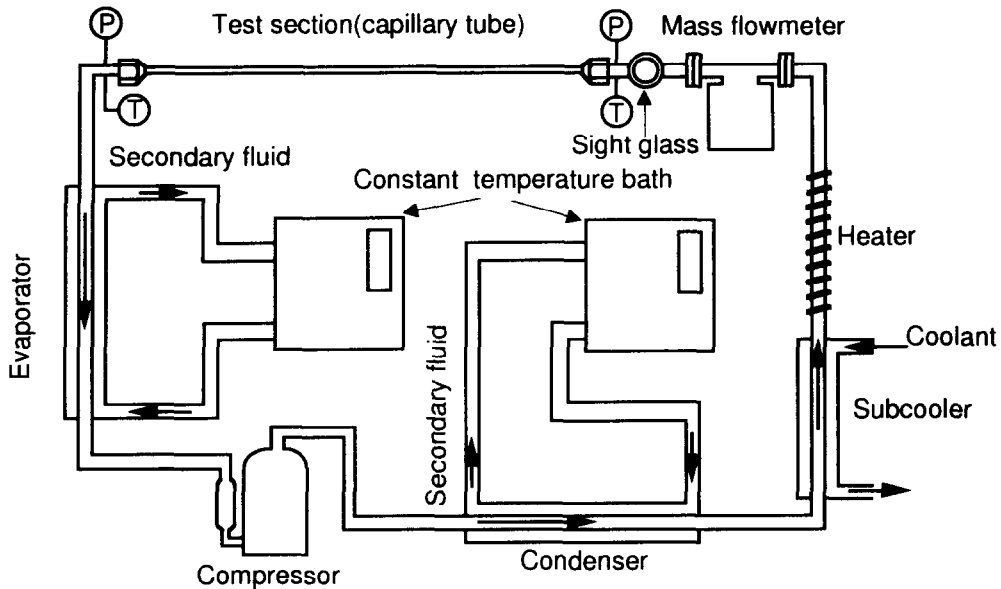


Figure 1. Schematic diagram of experimental apparatus.

parameter to calculate pressure drop of flows in capillary tubes was calculated by Haaland's equation (Haaland 1983) which consider the effect of roughness of capillary tube as well as Reynolds number, and these theoretical values were compared with experimental data. A model for the prediction of the frictional pressure drop of flows in capillary tubes is presented after comparing several homogeneous flow models with one another.

2. EXPERIMENTS

2.1. Experimental apparatus

The experimental apparatus is composed of condenser, compressor, evaporator and capillary tube as an expansion device. Those are the general components of refrigerating system. The schematic diagram is shown in figure 1. The condensation and evaporation pressure can be varied by adjusting the temperature and mass flow rate of water which is used as a secondary heat transfer fluid in the condenser and evaporator. The degree of subcooling is changed by using the subcooler

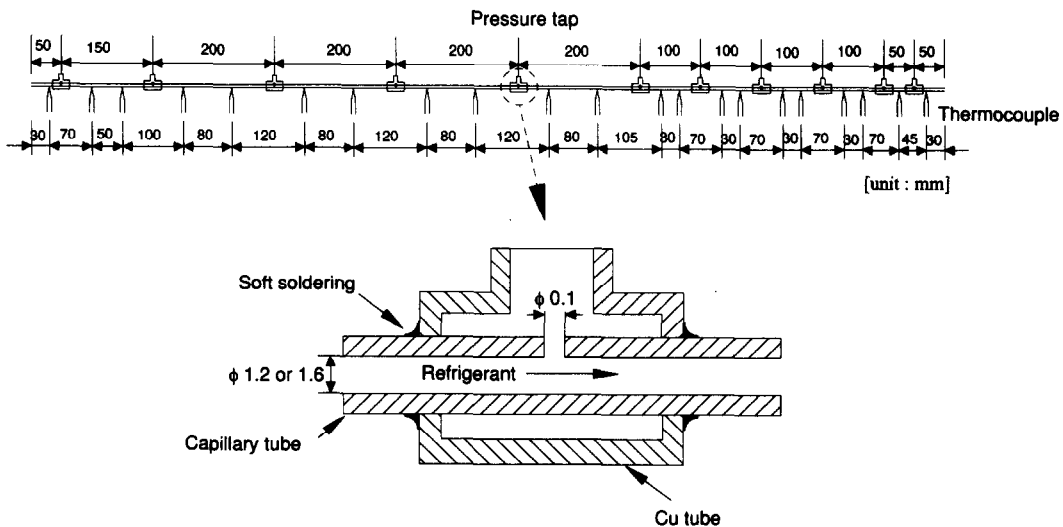


Figure 2. Overview of the tested capillary tube and the locations of pressure taps and thermocouples.

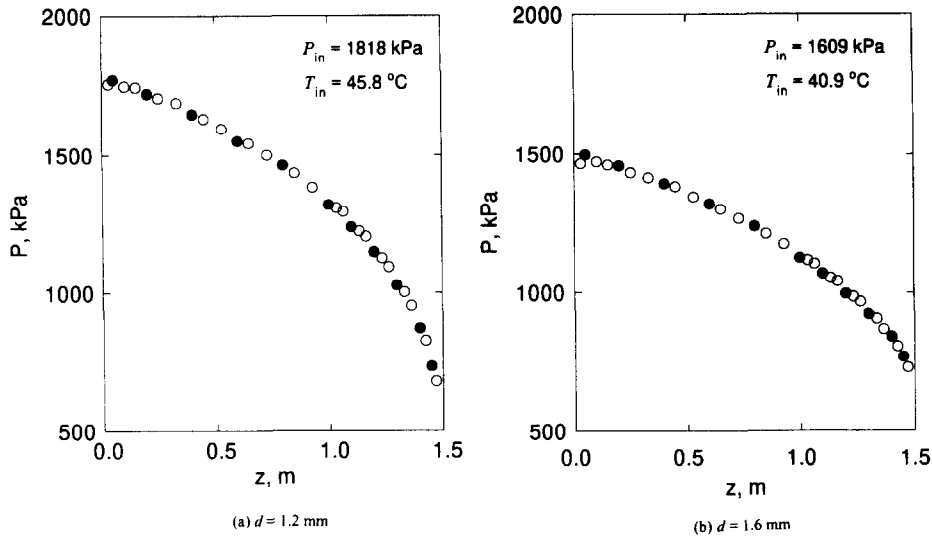


Figure 3. Distributions of the measured pressure and comparison with the saturation pressure. ●: $P_{\text{measurement}}$, ○: $P_{\text{saturation}}$. (a) $d = 1.2$ mm; (b) $d = 1.6$ mm.

and PID controlled electric heater. Mass flow meter (accuracy $\pm 0.48\%$) for measuring the mass flow rate of the refrigerants, pressure transducer (accuracy $\pm 0.2\%$) to measure inlet and outlet pressure of the capillary tube and the pressure distribution along the tube, the T-typed thermocouple for the temperature measurements, and hybrid recorder are used as measuring equipments.

When the thermocouple is inserted in the tube, it will affect the flow because of the slenderness of the capillary tube with the diameter of about 1 mm. Therefore, the outside wall temperature is measured with T-typed thermocouple and used to calculate the temperature of refrigerant in the capillary tube. For pressure measurement, the holes on the capillary tube were made by laser beams as shown in figure 2 to avoid any chips remaining inside the capillary tube. The diameter of these holes were about 0.1 mm. These holes were covered with copper tube whose inner diameter is larger than that of capillary tube and a pressure tap was made on the copper tube. Eleven points for pressure and 21 points for temperature measurement are shown in figure 2. The outside surface of the capillary tube was then insulated.

In order to check the validity of choosing the wall temperature as the refrigerant temperature, the measured pressure was compared with the saturation pressure calculated from the corresponding wall temperature. The inlet of the capillary tube was adjusted to a saturated liquid state or a subcooled liquid state with small degree of subcooling, and the pressure and temperature were measured along the capillary tube. Figure 3 shows the saturated pressure calculated from the measured wall temperature and measured pressure when R22 was used. The saturated pressure is calculated by modified Carnahan–Starling equation of state (Kim *et al.* 1991). Since the measured pressure is well fit for the saturated pressure within 1.5% error bound at the maximum, it is very good to choose the wall temperature as the temperature of refrigerant in the capillary tube.

2.2. Experimental conditions

The refrigerants used in this study are the pure HFC refrigerants such as R32, R125, R134a, and the binary refrigerant mixtures such as R32/R134a (30/70 by mass fraction), R32/R125 (60/40), R125/R134a (30/70), and ternary refrigerant mixture such as R32/R125/R134a (23/25/52). Two capillary tubes made of copper are used in this study, which are 1.5 m long with inside diameters of 1.2 and 1.6 mm, respectively. The ranges of pressure at inlet of capillary tube in the present experiment are 1025–2800 kPa, which correspond to condensation temperatures of 40–50°C, and the ranges of degree of subcooling of 1.8–12.2°C, respectively. Different refrigerants have different the inlet pressure of capillary tube, because experiments are done under the condition that the condensation temperature is constant. Table 1 shows the specification of capillary tubes and experimental conditions.

Table 1. Experimental conditions in this study

		I	II
Capillary tube	I.D. (mm)	1.2	1.6
	Length (m)	1.5	1.5
	Roughness (μm)	0.1963	0.6894
Inlet pressure (kPa)	1025–2800		
Outlet pressure	Critical condition		
Mass velocity ($\text{kg}/\text{m}^2 \text{ s}$)	3980–9370		
Degree of subcooling ($^{\circ}\text{C}$)	1.8–12.2		
Refrigerant	#1 R32		
	#2 R125		
	#3 R134a		
	#4 R32/R134a (30/70)		
	#5 R32/R125 (60/40)		
	#6 R125/R134a (30/70)		
	#7 R32/R125/R134a (23/25/52)		

3. FRICTION FACTORS OF LIQUID PHASE FLOW

The subcooled liquid region begins at the inlet of the capillary tube and ends at the point where vaporization starts. In this region, the refrigerant can be assumed to be an incompressible fluid, so pressure drop is only caused by friction with the wall of capillary tube.

The friction factor of liquid phase flow, f is

$$f = -2 \frac{\rho_L d}{\dot{m}^2} \frac{dP}{dz}, \quad [1]$$

where f , ρ_L , d , \dot{m} , P , and z are the Darcy friction factor, the density of liquid phase, internal diameter of capillary tube, mass velocity, pressure, and axial length, respectively. In this study, the density of the liquid phase is calculated by the modified Carnahan–Starling equation of state (Kim *et al.* 1991). Therefore, the friction factor of liquid phase flow can be obtained from the mass velocity, refrigerant density and pressure gradient calculated from the measured pressure.

The surface roughness as well as Reynolds number influence the friction factor of the flow in the capillary tube (Lin *et al.* 1991). The diameter of the capillary tube is so small that the relative roughness has a remarkable effect on the shear stress at the wall, even though the absolute roughness is small. In order to include the relative roughness, Haaland's equation (Haaland 1983) for the determination of the friction factor is used

$$f = \left\{ -1.8 \log \left[\frac{6.9}{\text{Re}} + \left(\frac{\epsilon}{3.7d} \right)^{1.11} \right] \right\}^{-2}, \quad [2]$$

$$\text{Re} = \frac{\dot{m}d}{\eta}, \quad [3]$$

where η is the viscosity and ϵ is the wall roughness. The data from Ripple & Matar (1993) and Olieveria & Wakeham (1993a) were used for the viscosity of the liquid of the pure refrigerant to calculate Reynolds number.

The wall roughnesses of capillary tubes in this study are $0.1963 \mu\text{m}$ for $d = 1.2 \text{ mm}$ and $0.6894 \mu\text{m}$ for $d = 1.6 \text{ mm}$ as shown in table 1. Because of this roughness, the friction factor can be increased by 4.6% for 1.2 mm or 12% for 1.6 mm tube at the maximum. Figure 4 shows the friction factor calculated by [2], in comparison with the experimental data. The solid lines represent the theoretical value from Haaland's equation, and the symbols the experimental value obtained by [1]. The relative root mean square error between the theoretical and experimental value is 7.6%.

For the refrigerant mixture, the viscosity of the refrigerant mixture is needed to obtain the Reynolds number described by [3]. So the mixing rule for the viscosity of refrigerant mixture is required. Among various mixing rules, equation [4] suggested by Grunberg & Nissan (1949) is

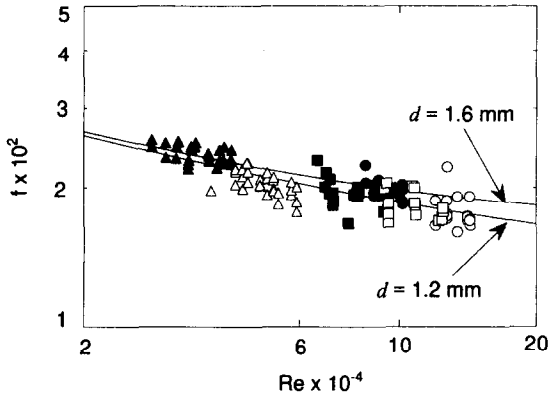


Figure 4. Comparison of friction factor calculated from Haaland's equation (solid line) with the experimental data for pure refrigerants. ●: R32, $d = 1.2$ mm; ○: R32, $d = 1.6$ mm; ■: R125, $d = 1.2$ mm; □: R125, $d = 1.6$ mm; ▲: R134a, $d = 1.2$ mm; △: R134a, $d = 1.6$ mm.

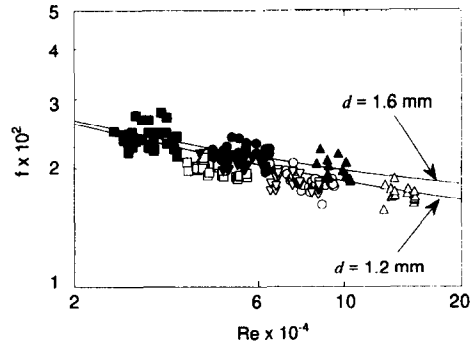


Figure 5. Comparison of friction factor calculated from Haaland's equation (solid line) with the experimental data for refrigerant mixtures. ●: R32/R134a, $d = 1.2$ mm; ○: R32/R134a, $d = 1.6$ mm; ■: R32/R125, $d = 1.2$ mm; □: R32/R125, $d = 1.6$ mm; ▲: R125/R134a, $d = 1.2$ mm; ▼: R125/R134a, $d = 1.6$ mm; ▲: R32/R125/R134a, $d = 1.2$ mm; ▼: R32/R125/R134a, $d = 1.6$ mm.

known to be a simple form and applied in various mixtures.

$$\ln \eta_m = \sum_i x_i \ln \eta_i + \sum_{i \neq j} x_i x_j G_{ij}, \quad [4]$$

where η_m is the viscosity of the mixture, η_i is the viscosity of i component, x_i and x_j are the mole fraction of i component and j component of the liquid phase respectively, and G_{ij} is the binary interaction parameter.

The binary interaction parameter of refrigerant mixtures was calculated based upon the experimental data. The binary interaction parameter is assumed constant and calculated by the experimental data (Bivens *et al.* 1993). The G_{ij} s of R32/R134a, R32/R125 and R125/R134a are represented in table 2. The G_{ij} of R125/R134a is calculated by equation [4] because there are no experimental data of viscosities of R125/R134a.

Figure 5 presents the friction factor of the refrigerant mixtures calculated by [2] and G_{ij} of table 2, in comparison with the experimental data. The relative root mean square error of 8.75% is in the error bound of the pure refrigerant, so Haaland's equation with Grunberg & Nissan's mixing rules can be used in case of the refrigerant mixtures as well as pure refrigerants if G_{ij} of mixtures is known.

4. PRESSURE DROP OF TWO-PHASE FLOW

4.1. Causes of pressure drop of a capillary tube

The pressure drop in the two-phase flow region is caused by the friction at the wall surface of capillary tube and the acceleration due to the increase of specific volume as shown in equation [5]

$$-\left(\frac{dP}{dz}\right)_{t, total} = f_t \frac{\dot{m}^2 v_t}{2d} + \dot{m}^2 \frac{dv_t}{dz}, \quad [5]$$

where the two-phase friction factor, f_t can be obtained by using the average viscosity of the two-phase region, η_t , and v_t is the specific volume of the two-phase region. These two contributions

Table 2. Binary interaction parameter for the calculation of the viscosity of refrigerant mixtures

	G_{ij}	Data source for η_m
R32/R134a (25/75)	0.0795	Bivens <i>et al.</i> (1993)
R32/R125 (60/40)	-0.2	Bivens <i>et al.</i> (1993)
R125/R134a (30/70)	1.6215†	Not available
R32/R125/R134a (30/10/60)	—	Bivens <i>et al.</i> (1993)

†Calculated from the viscosity of R32/R125/R134a.

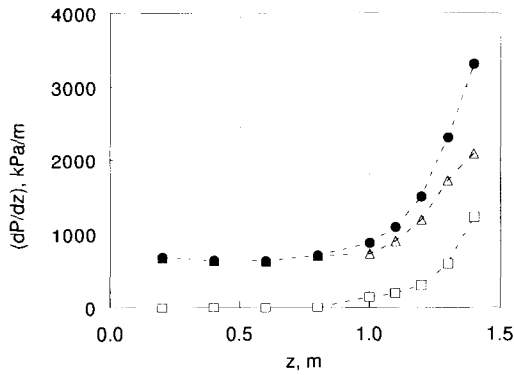


Figure 6. Pressure gradients of R32 along capillary tube with $d = 1.2$ mm, $P_{in} = 2792$ kPa, $\Delta T_{sub} = 10.7^\circ\text{C}$. \bullet : $(dP/dz)_{total}$, \triangle : $(dP/dz)_{friction}$, \square : $(dP/dz)_{acceleration}$.

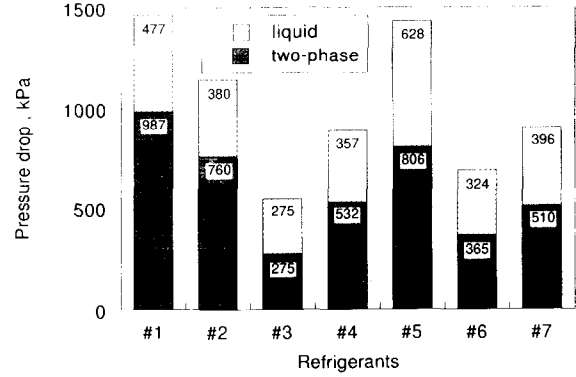


Figure 7. Pressure drops of liquid and two-phase region along capillary tube of various refrigerants at $T_{cond} = 45^\circ\text{C}$, $d = 1.2$ mm and $\Delta T_{sub} \approx 11.2^\circ\text{C}$. #1: R32, #2: R125, #3: R134a, #4: R32/R134a, #5: R32/R125, #6: R125/R134a, #7: R32/R125/R134a.

to the pressure drop amplify each other, so that the pressure decreases more and more as the fluid approaches the exit of the capillary tube. Figure 6 shows pressure gradient of R32 along the capillary tube with $d = 1.2$ mm, $P_{in} = 2792$ kPa, and $\Delta T_{sub} = 10.7^\circ\text{C}$. $(dP/dz)_{friction}$ is the first term and $(dP/dz)_{acceleration}$ is the second term of right hand side of [5]. There is only frictional pressure drop in the liquid flow region, but there are frictional and accelerational pressure drops in the two-phase flow region. The pressure drop of the two-phase region is much larger than that of the liquid region, and the frictional pressure drop dominates as shown in figure 6.

The pressure drops from the first pressure tap ($z = 0.05$ m) to the last pressure tap ($z = 1.45$ m) along capillary tube with $d = 1.2$ mm, $T_{cond} = 45^\circ\text{C}$, and $\Delta T_{sub} \approx 11.2^\circ\text{C}$ are shown in figure 7. This pressure drop excludes the one caused by sudden contraction at the inlet of the capillary tube and sudden enlargement at the outlet of the capillary tube. In this study, condensation temperature, T_{cond} , is maintained constant to compare with various refrigerants, so different refrigerants have different pressures at the inlet of the capillary tube because saturation pressures corresponding to condensation temperatures are not the same. Pressure drop increases as saturation pressure of refrigerants increases, and pressure drop of R32 is the largest and that of R134a is the smallest. The pressure drop of R32/R134a (#4) is about equal to that of R32/R125/R134a (#7). The pressure drop of the two-phase region is larger than that of the liquid region though the length of the two-phase region is shorter than that of the liquid region when the degree of subcooling is large as 11.2°C . So, the pressure drop of the two-phase region is very important to analyze the flow of capillary tube. But it is not simple to separate subcooled liquid region and two-phase region of refrigerant flow in capillary tube. For vaporization starts when the pressure is equal to the saturation pressure corresponding to inlet temperature theoretically, but vaporization will occur below saturation pressure. This difference of saturation pressure from vaporization pressure is called underpressure for vaporization. So, it is not easy to know when vaporization of refrigerant flow in capillary tube starts, and a correlation for calculating the real position of the transitional point from the subcooled liquid region to the two-phase flow region is needed. An empirical correlation for underpressure was developed as equation [6]

$$P_{sat} - P_{vap} = c_1 + c_2 \times \dot{m} + c_3 \times \Delta T_{sub} + c_4 \times \dot{m} \times \Delta T_{sub}, \quad [6]$$

where $P_{sat} - P_{vap}$ is the underpressure for vaporization with unit of kPa, \dot{m} is mass velocity with unit of $\text{kg/m}^2\text{s}$, ΔT_{sub} is degree of subcooling with unit of $^\circ\text{C}$, and coefficients c_1 , c_2 , c_3 , c_4 are calculated from experimental data by the least squares method and represented in table 3. The validity of [6] is limited to the mass velocity and the degree of subcooling range as shown in table 1.

4.2. Two-phase frictional multiplier

It is reasonable to utilize the homogeneous flow model in two-phase flow region since the two-phase flow in the capillary tube is a bubbly flow region in the high liquid flow range of this

Table 3. Coefficients of correlation for underpressure

Refrigerants	$c_1 \times 10^{-2}$	$c_2 \times 10^2$	$c_3 \times 10^{-1}$	$c_4 \times 10^3$
R32	-2.855164	5.451684	3.548410	-4.707993
R125	-0.862453	1.573368	4.015019	-4.381408
R134a	-1.412876	5.457141	0.974871	-3.950501
R32/R134a (30/70)	-3.911112	7.427592	4.705004	-8.561734
R32/R125 (60/40)	-3.173929	4.692433	4.920039	-5.463006
R125/R134a (30/70)	-6.986621†	0.558221	-0.501419	0.502433
R32/R125/R134a (23/25/52)	-1.536972	3.844115	1.818622	-3.036465

† $c_1 \times 10^2$.

study. The pressure drop caused by the friction can be presented as equation [7]

$$-\left(\frac{dP}{dz}\right)_{t,friction} = f_t \frac{\dot{m}^2 v_t}{2d} = f_t \frac{\dot{m}^2 v_L}{2d} \left[1 + x \frac{(v_G - v_L)}{v_L} \right], \tag{7}$$

where v_L and v_G are the specific volume of saturated liquid and vapor; and the two-phase friction factor, f_t can be obtained by using the average viscosity of the two-phase region, η_t , and quality, x can be calculated by the energy equation. Because the capillary tube in the experiment is insulated, the flow of refrigerant is considered as adiabatic. The energy equation in the two-phase flow region is written as

$$h_0 = h_t + \frac{(\dot{m}v_t)^2}{2} = [h_L + x(h_G - h_L)] + \frac{\dot{m}^2[v_L + x(v_G - v_L)]^2}{2} = \text{constant}, \tag{8}$$

where h_0 is the total specific enthalpy, h_t is the specific enthalpy of the two-phase region, and h_L and h_G are the specific enthalpy of saturated liquid and vapor. The specific enthalpy and specific volume at saturation state are easily calculated by the measured temperature and pressure, and quality is only unknown variable in [8].

The pressure drop due to the friction of the two-phase flow can be represented as a function of the pressure drop of single liquid flow and the two-phase frictional multiplier (Collier & Thome 1994) as shown in equation [9]

$$-\left(\frac{dP}{dz}\right)_{t,friction} = -\left(\frac{dP}{dz}\right)_{lo} \phi_{lo}^2, \tag{9}$$

where ϕ_{lo}^2 is the two-phase frictional multiplier and the subscript lo implies liquid phase flow only.

Therefore, $-(dP/dz)_{lo}$ is the frictional pressure gradient calculated from [7] for the total flow assumed as liquid one by substituting f_{lo} , v_L for f_t , v_t at [7], respectively. The left-hand side of [9], the frictional pressure gradient at the two-phase region can be calculated with measured pressure data and the distance between pressure tap and the next one. So, the two-phase frictional multiplier can be calculated with experimental data of pressure, mass velocity etc. by [9]. Figure 8 shows the two-phase frictional multipliers of the pure refrigerants, R32, R125, and R134a. The two-phase frictional multiplier increases as the quality increases and it converges to 1 as the quality approaches zero because the flow pattern becomes similar to a single-phase liquid flow. For the same quality, the two-phase frictional multiplier of R125 is smaller than that of R32 and R134a. The main reason is that the tube inlet pressure differs in various refrigerants when keeping the condensation temperature constant. While the reduced pressure comes within the range of 0.559–0.707 for R125, that of R32 or R134a falls within the range of 0.423–0.477 or 0.251–0.325, respectively. Since $(v_G - v_L)/v_L$ grows smaller as the reduced pressure becomes larger, R125 with the higher reduced pressure will have the lower value of two-phase frictional multiplier.

Substituting [3] and [7] into [9], the two-phase frictional multiplier is obtained as follows.

$$\phi_{lo}^2 = \frac{f_t}{f_{lo}} \left[1 + x \left(\frac{v_G - v_L}{v_L} \right) \right]. \tag{10}$$

If we express f as a function of Reynolds number and the wall roughness by using [2], two-phase frictional multiplier can be written as follows

$$\phi_{lo}^2 = \left\{ \frac{\log \left[\frac{6.9}{Re_{lo}} + \left(\frac{\epsilon/d}{3.7} \right)^{1.11} \right]}{\log \left[\frac{6.9}{Re_t} + \left(\frac{\epsilon/d}{3.7} \right)^{1.11} \right]} \right\}^2 \left[1 + x \left(\frac{v_G - v_L}{v_L} \right) \right]. \tag{11}$$

To calculate two-phase frictional multiplier by [11], the two-phase region viscosity, η_t must be estimated from the viscosity of saturated liquid and vapor and quality. There are various equations to estimate η_t to homogeneous flow model as shown in equations [12]–[15]. These equations are suggested by McAdams *et al.* (1942), Cicchitti *et al.* (1960), Dukler *et al.* (1964), and Davidson *et al.* (1943), respectively.

$$\frac{1}{\eta_t} = \frac{x}{\eta_G} + \frac{(1-x)}{\eta_L}, \tag{12}$$

$$\eta_t = x\eta_G + (1-x)\eta_L, \tag{13}$$

$$\eta_t = \frac{[xv_G\eta_G + (1-x)v_L\eta_L]}{[xv_G + (1-x)v_L]}, \tag{14}$$

$$\eta_t = \eta_L \left[1 + x \left(\frac{v_G - v_L}{v_L} \right) \right], \tag{15}$$

where η_G and η_L is the viscosity of saturated vapor and liquid, respectively. Oliveira & Wakeham (1993b) and ASHRAE (1993) data are used for the viscosity of the saturated vapor of the pure refrigerant.

Figure 9 shows the calculated two-phase frictional multiplier of the pure refrigerants by several equations with the experimental values to figure out the best fitted equation in applying to the

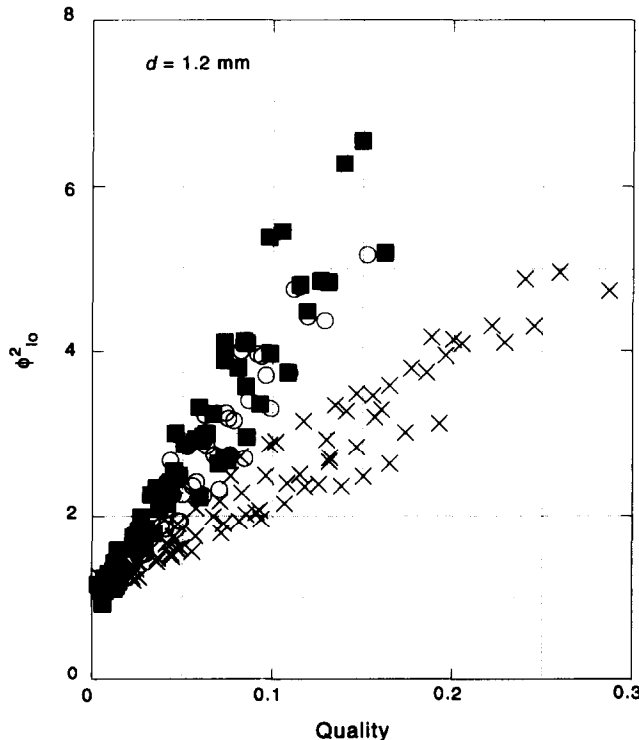


Figure 8. Variation of two-phase frictional multiplier with respect to quality. ○: R32, ×: R125, ■: R134a.

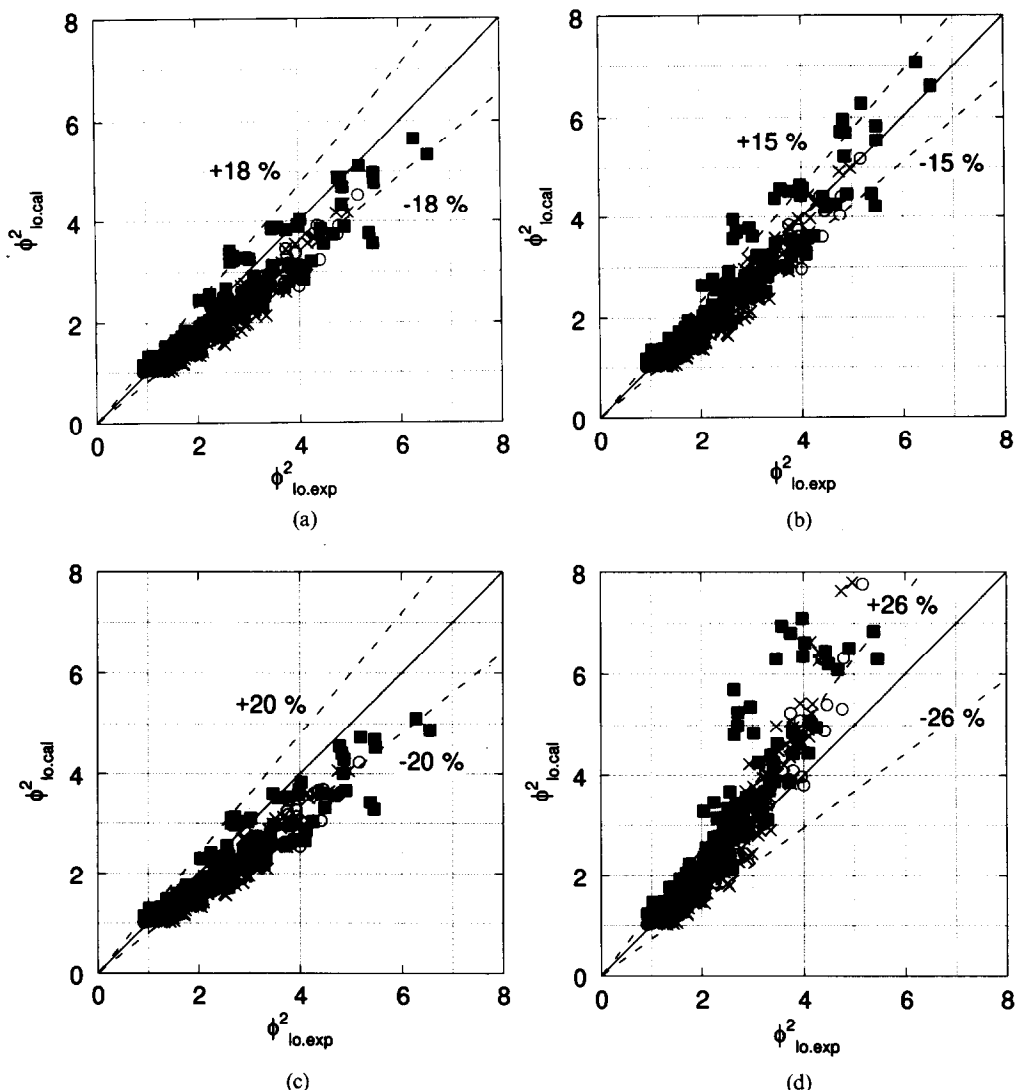


Figure 9. Comparison of the two-phase frictional multiplier calculated from several models with the experimental data for pure refrigerants. ○: R32, ×: R125, ■: R134a. (a) McAdams *et al.*; (b) Cicchitti *et al.*; (c) Dukler *et al.*; (d) Davidson *et al.*

capillary tube flow. The equation suggested by McAdams *et al.* (1942) and Dukler *et al.* (1964) present the value lower than that from experiment with relative root mean square errors of 18.1 and 19.8%, respectively. With the equation by Davidson *et al.* (1943), much higher value is estimated than the experimental data with relative root mean square error of 26.2%. The equation suggested by Cicchitti *et al.* (1960) is said to be the best fitted with the experimental value with relative root mean square error of 15.0%. Therefore, it is recommended to use Cicchitti *et al.*'s equation (1960) for estimating viscosities of two-phase region in the capillary tube.

For the refrigerant mixtures, the viscosity of the liquid phase is estimated by using the mixing rule suggested by Grunberg & Nissan (1949) and the binary interaction parameter in Table 2. For the viscosity of the vapor phase of the refrigerant mixtures, the mixing rule suggested by Wilke (1950) is used

$$\eta_m = \frac{\sum_{i=1}^n y_i \eta_i}{\sum_{j=1}^n y_j \phi_{ij}}, \tag{16}$$

$$\phi_{ij} = \left(\frac{M_j}{M_i} \right)^{1/2} = \phi_{ji}^{-1}, \tag{17}$$

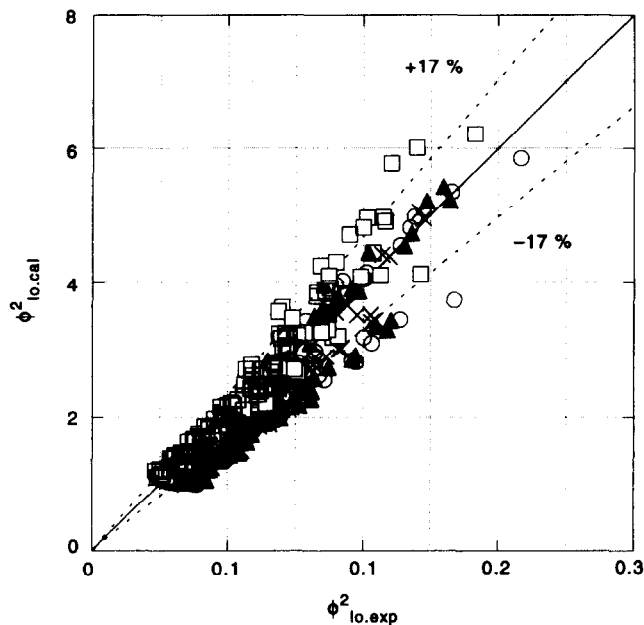


Figure 10. Comparison of the two-phase frictional multiplier calculated from Cicchitti's equation with the experimental data for refrigerant mixtures. \circ : R32/R134a, \times : R32/R125, \blacktriangle : R125/R134a, \square : R32/R125/R134a.

where x_i and x_j are the mole fractions of i component and j component of the vapor phase, and M_i and M_j are the molecular weights of i component and j component, respectively. The viscosity of the two-phase region is calculated by [13] suggested by Cicchitti *et al.* (1960) which was used for the calculation of viscosity of the pure refrigerant in the two-phase. Figure 10 shows the two-phase frictional multiplier of the refrigerant mixtures with the experimental values. The relative error is 17.1%.

5. CONCLUSIONS

The pressure drops of HFC refrigerants such as R32, R125, R134a, and their mixtures such as R32/R134a (30/70), R32/R125 (60/40), R125/R134a (30/70), R32/R125/R134a (23/25/52) in capillary tube are investigated experimentally.

In this study, the range of Reynolds number of refrigerant flow in capillary tubes is 2×10^4 to 2×10^5 . In this range, roughness of the tube wall as well as Reynolds number affects friction factor of flows. So theoretical equation for estimating the friction factor of refrigerant flow in capillary tubes has to describe the friction factor as a function of Reynolds number and the roughness of the tube wall. The friction factors of the refrigerants flow in a capillary tube are estimated by Haaland's equation. Experimental data of the pure refrigerants flows for single-phase liquid agree with this equation within a relative error of about 7.6%. The viscosities of refrigerant mixtures in the liquid state are estimated by the mixing rule suggested by Grunberg & Nissan (1949) and the binary interaction parameter. Relative errors of experimental friction factor and calculated value of refrigerant mixtures flows are about 8.75%. If we know the binary interaction parameter of refrigerant mixtures, we can calculate the friction factor of refrigerant mixture flowing in a capillary tubes.

Pressure drop increases as saturation pressure of refrigerants increases in case of condensation temperature being maintained constant. The pressure drop of R32 along the capillary tube is the largest and that of R134a is the smallest among refrigerants we used. The pressure drop of two-phase region is larger than that of liquid region, so the pressure drop of two-phase region is very important. The two-phase frictional multiplier for the calculation of frictional pressure drops at liquid-vapor two-phase flow is predicted by several homogeneous flow models. In the high liquid flow range such as experimental range of this study, the homogeneous flow model is appropriate

to calculate the pressure drop of two-phase flow in capillary tubes. Results show that Cicchitti's equation (Cicchitti *et al.* 1960) is the most adequate for adiabatic capillary flows.

The results of this study can be used to estimate the pressure drop of the refrigerant mixtures flowing in the capillary tube.

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