Flow and Pressure Drop Characteristics of R22 in Adiabatic Capillary Tubes

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The objective of this study is to present flow and pressure drop characteristics of R22 in adiabatic capillary tubes of inner diameters of 1.2 to 2.0 mm, and tube lengths of 500 to 2000 mm. Distributions of temperature and pressure along capillary tubes and the refrigerant flow rates through the tubes were measured for several condensing temperatures and various degrees of subcooling at the capillary tube inlet. Condensing temperatures of R22 were selected as 40, 45, and 50℃ at the capillary tube inlet, and the degree of subcooling was adjusted to 1 to 18℃. Experimental results including mass flow rates and pressure drops of R22 in capillary tubes were provided. A new correlation based on Buckingham Π theorem to predict the mass flow rate through the capillary tube was presented considering major parameters which affect the flow and pressure drop characteristics.

Key Words: Alternative Refrigerant, Capillary Tube, Mass Flow Rate, Pressure Drop, R22

Nomenclature

A : Constant defined in Eq. (3)
a : Exponent defined in Eq. (3)
b : Exponent defined in Eq. (3)
c : Exponent defined in Eq. (3)
c_P : Specific heat at inlet of capillary tube, J/(kg·K)
d : Inner diameter of capillary tube, mm
f : Friction factor
G : Mass flux, kg/(m^2 · s)
L : length of capillary tube, mm
$\dot{m}$: Mass flow rate, kg/s
P : Pressure, kPa
P_{sat} : Saturation pressure corresponding to the measured refrigerant temperature, kPa
P_{meas} : Measured pressure of refrigerant, kPa
P_s : Saturated state pressure, kPa
P_v : Measured pressure of refrigerant at the onset point of vaporization, kPa
P_{in} : Inlet pressure of capillary tube, kPa
T_{sub} : Degree of subcooling, ℃
v : Specific volume at the inlet of capillary tube, m^3/kg
$\bar{v}$: Average of specific volume, m^3/kg
x : Quality
z : Length of capillary tube, mm

Greek Symbols

α : Void fraction
μ : Viscosity at the inlet of capillary tube, kg/(m·s)
ρ : Density, kg/m^3

Subscripts

a : Acceleration
f : Saturated liquid phase
g : Saturated vapor phase

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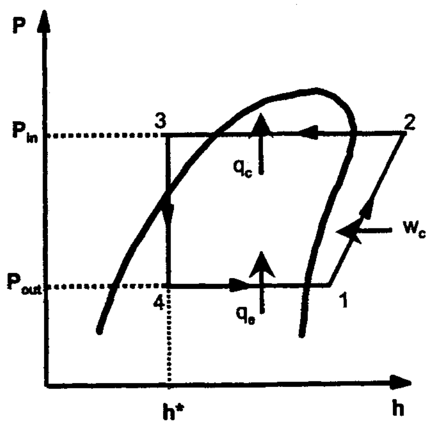
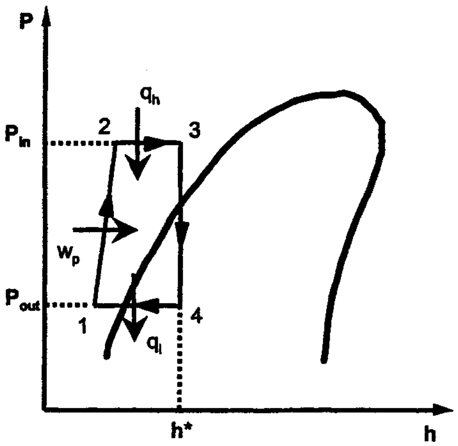
(Manuscript **Received** November 24, 2000; **Revised** July 3, 2001)

1. Introduction

Capillary tube is widely used as an expansion device in small refrigeration and air-conditioning



(a) Compressor-driven system



(b) Pump-driven system

Fig. 1 Pressure-enthalpy diagram of refrigerant in a capillary tube test loop

systems. It is a simple tube with inner diameters of a few millimeters, but the flow and pressure drop inside a capillary tube are very complex and have a strong influence on the performance of the system. The selection of proper diameter and length of a capillary tube for a specific application is made mostly by a trial-and-error procedure. In order to achieve designed optimal performance, it is required to understand the flow and pressure drop characteristics and to choose an appropriate capillary tube which fits to the system.

Refrigerant flow through capillary tubes is characterized by frictional pressure drop of two phase flow, mass flow rate, critical flow rate, and underpressure of vaporization. These characteristics of capillary tubes have been studied experimentally or analytically by many researchers (Bolstad and Jordan, 1948; Cooper et al., 1957; Koizumi and Yokoyama, 1980; Chen et al., 1990; Kuehl and Goldschmidt, 1990; 1991; Li et al., 1990; Wijaya, 1992; Melo et al., 1994; 1995; Chang and Ro, 1996a; 1996b; Kim et al., 1997; Bittle et al. 1998; Kim and Choi, 1998; and Chen et al., 2000). Many researchers have carried out experiments or analyses on pure refrigerants such as R12, R134a as an alternative to R12 and R22. However, test data, model predictions or correlations for flow characteristics reported by many researchers do not match well.

The major objective of this study is to provide a set of capillary tube performance data for R22. In this paper, pressure and temperature change in the capillary tubes are provided and the measured mass flow rate and underpressure of vaporization for several inlet conditions of different capillary tube geometries are also given for R22.

2. Experiments

2.1 Experimental apparatus

In our experiment, the refrigerant flow loop is designed to be driven by a magnetic gear pump in order to remove the effect of lubrication oil. Pressure-enthalpy diagram is shown in Fig. 1 to compare the test cycle using a magnetic gear pump with a vapor compression cycle using a compressor. As seen in this figure, the thermodynamic states of the refrigerant at the capillary tube inlet (point 3) and during the expansion process (process 3-4) were kept the same as for a vapor compression cycle of air-conditioners. By keeping the same inlet conditions to capillary tubes, the behavior of refrigerant in capillary tubes in this study is very similar to that of an actual vapor compression cycle. When a magnetic gear pump is used, the effect of lubrication oil in the compressor driven system can be effectively removed, and thus the complexity with oil such as the change in thermodynamic properties of refrigerant and oil mixtures, the formation of oil film in the capillary

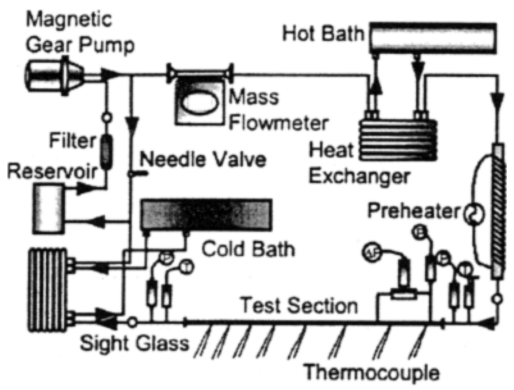


Fig. 2 Schematic diagram of experimental apparatus for capillary tube performance test

tubes, and the influence of oil on the flashing behavior of refrigerant can be avoided. The test in this study was carried out with no lubrication oil.

The test apparatus to investigate flow and pressure drop characteristics of various capillary tubes for R22 is shown in Fig. 2. The test setup consisted of magnetic gear pump, two brazed plate heat exchangers, mass flowmeter, refrigerant reservoir, preheating system, and test section with a capillary tube. Cold bath was installed to subcool the refrigerant after being flashed in the capillary tube.

By changing the speed of the magnetic gear pump, mass flow rate of refrigerant was varied. Since one of the important parameters in this study was the inlet pressure of the capillary tube, the temperature of the secondary fluid entering the heat exchanger from the hot bath was adjusted to control the inlet pressure of the test section. Preheater was also installed for precise control of inlet pressure using electricity.

The mass flow rate of refrigerant was measured by a mass flowmeter in the liquid line between the magnetic gear pump and the hot heat exchanger. The pressures of the inlet and the outlet of the test section were measured by absolute pressure transducers and the pressures along the test section were measured by differential pressure transducers considering the inlet pressure as a reference. Sight glasses were installed just before and after the test section for visual confirmation of the flow condition.

The small holes of diameter of about 0.1 mm on the capillary tubes were made by a laser beam for pressure measurement. The temperatures of the refrigerant along the capillary tube were measured by T-type thermocouples, which were soldered on the outside of the capillary tube. The whole capillary tube was insulated by circular foaming material with outer diameter of about 5 cm. The measured outside wall temperature can be considered as the temperature of refrigerant when the radial and circumferential heat conduction through the tube wall was neglected.

Experimental uncertainties were estimated as 0.5℃ for temperature measurement, 0.25% for pressure measurement, and 0.42% for the measurement of mass flow rates in the capillary tubes.

2.2 Experimental conditions

The major refrigerant used in this study was R22, and several capillary tubes made of copper were used in this study. For R22, five different capillary tubes with an inner diameter of 1.2, 1.3, 1.5, 1.7 and 2.0 mm (roughness of 0.12, 0.10, 0.09, 0.09, and 0.15 μm, respectively) were tested for pre-determined length of 500, 1000, 1500, and 2000 mm. The capillary tube inlet pressures were selected which corresponded to the saturation temperature in the condenser of air-conditioners. The condensing temperatures were selected as 40, 45, and 50℃, and the degree of subcooling, calculated from the saturated liquid temperature, was adjusted to 1 to 18℃. Thermodynamic properties of refrigerants were obtained using REFPROP (Huber et al., 1996).

3. Experimental Results and Discussion

3.1 Refrigerant's state change

The thermodynamic state of refrigerant in a capillary tube can be estimated from measured pressures and temperatures. If the measured pressure is higher than the saturation pressure corresponding to the measured temperature, the refrigerant will be in a liquid phase (region A in Fig. 3). In the capillary tube, there exists a region

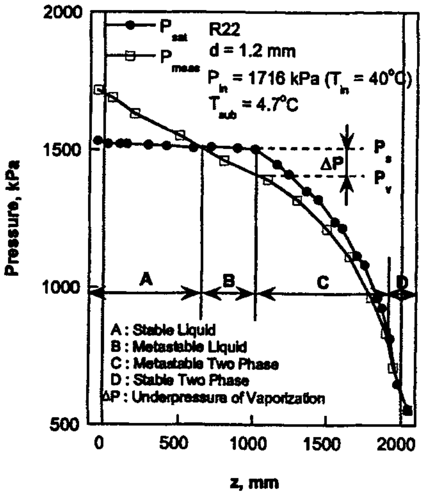


Fig. 3 Distribution of measured pressure and saturation pressure corresponding to the measured temperature along the capillary tube

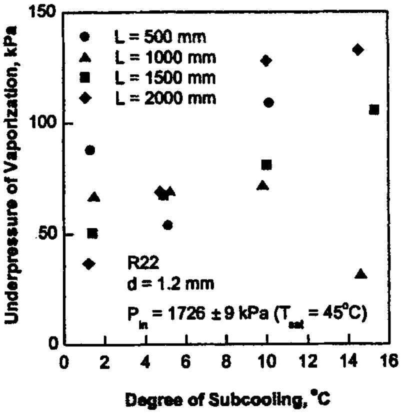


Fig. 4 Underpressure of vaporization for different the degree of subcoolings at the condensing temperature of 45℃

where the refrigerant temperature is not changing much by maintaining a similar temperature as in the upstream, while the measured pressure is continuously decreasing. In this region the measured pressure is lower than the calculated saturation pressure corresponding to the measured temperature. This, in turn, implies that the saturation temperature of refrigerant for the measured saturation pressure is lower than the measured temperature of the refrigerant (region B in Fig. 3). At the downstream next to it, the calculated saturation pressure is being lowered, and there are two regions; one is where the measured pressure is lower than the saturation pressure calculated based on the measured refrigerant temperature (region C in Fig. 3), and the other is where both the pressures are the same (region D in Fig. 3). The two phase flow behavior shown in Fig. 3, was observed in almost every case. In the region where the measured pressure is lower than the calculated saturation pressure based on the refrigerant temperature measurement, but the refrigerant exists as liquid phase, it is believed that the refrigerant is in a metastable liquid state. The difference between the measured pressure of refrigerant at the onset point of vaporization and the saturation pressure corresponding to the measured refrigerant temperature, which is $P_s - P_v$ as shown in Fig. 3, is designated as the underpressure of vaporization. This underpressure of vaporization exists due to the fact that a finite amount of pressure difference is required for the onset of vaporization. If the flow in capillary tube is estimated without considering the underpressure of vaporization, the capillary tube length for the liquid region is estimated to be shorter than the actual length. Therefore, the existence of the metastable liquid region results in a higher mass flow rate than that would exist under an ideal thermodynamic equilibrium condition.

Figure 4 shows the underpressure of vaporization determined from experimental data as a function of degree of subcooling. In the case of a high degree of subcooling, the saturation point of the refrigerant moves toward the downstraem of the capillary tube where there is a higher pressure fluctuation due to turbulent characteristics of the flow, therefore, a decrease of the underpressure of vaporization is expected. However, as stated by Li et al. (1990), an increase of mass flow rate results in an increase of the underpressure of vaporization, because both the frictional pressure drop and the flow velocity which influences the rate of depressurization increase with increased mass flow rate. It is known that underpressure of vaporization

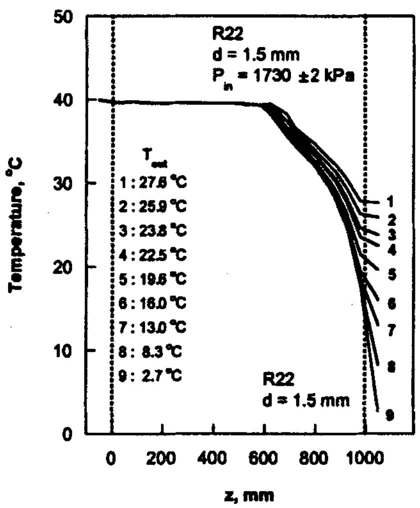


Fig. 5 Measured temperature distribution along the capillary tube with various outlet temperatures

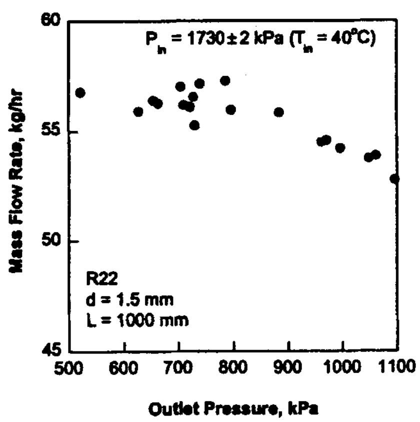


Fig. 6 Effect of capillary tube outlet pressure on the mass flow rate through the tube

increases as the mass flux of refrigerant increases and the degree of subcooling decreases (Li et al., 1990). The underpressure of vaporization is influenced by two factors : pressure fluctuation and the rate of depressurization. The former has a negative effect on the underpressure of vaporization, while the latter has a positive effect. On the other hand, Meyer and Dunn (1998) stated that if a pressure fluctuation increases the local pressures at the nucleation site, the nucleation site may be extinguished, thus metastable region will increase.

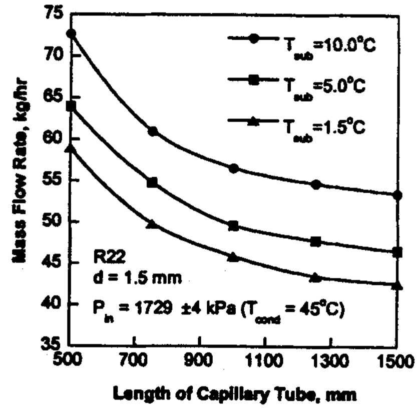


Fig. 7 Variation of mass flow rate versus length of a capillary tube for several degrees of subcooling

It is also stated that the higher rate of depressurization will result in shorter metastable lengths, which is quite contrary to Li et al. 's results. In this experiment the mass flow rate increased with higher degree of subcooling (see Fig. 7), therefore, the experimental data shown in Fig. 4 are not for the same mass flow rates through the capillary tube, which is quite a normal situation in the test for capillary tubes. Hence, the results shown in Fig. 4 indicate that the underpressure of vaporization can increase or decrease depending on the opposite contributions of increased mass flow rate and increased degree of subcooling.

3.2 Mass flow rate through capillary tubes

The temperature change along the capillary tube for a fixed inlet pressure with different outlet temperatures (or corresponding saturation pressure) is shown in Fig. 5. As shown in Fig. 5, even though the outlet temperature (or corresponding outlet pressure in two phase) of the capillary tube is changing for a fixed inlet pressure, liquid phase region does not actually change, where the measured temperature of the refrigerant is nearly constant. Also, at outlet temperatures below a certain limit, it is found that the temperature change in the capillary tube is quite the same for liquid and two phase regions (lines 6 to 9 in Fig. 5). This implies that the effect

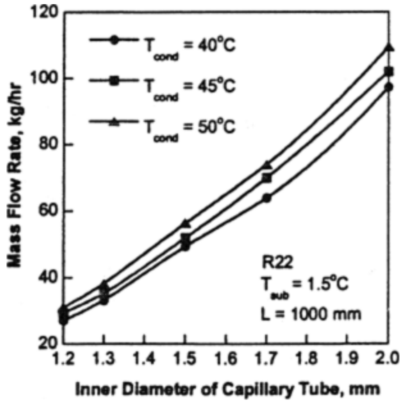


Fig. 8 Variation of mass flow rate versus inner diameter of the capillary tube for several condensing temperatures

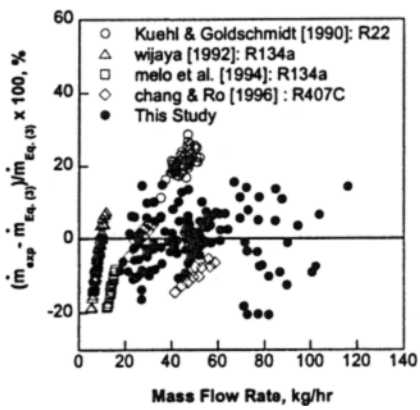


Fig. 9 Comparison of experimental data with mass flow rates calculated from Eq. (3)

of outlet pressure below a certain limit on the mass flow rate of refrigerant is minor, that is, the experimental condition is a choking condition.

The mass flow rate for a fixed inlet pressure with different outlet pressures is shown in Fig. 6. This figure is closely connected with Fig. 5. As for the outlet pressure below about 800 kPa (corresponding saturation temperature of 15. 5℃), the mass flow rate through the capillary tube remains quite constant. Therefore, parameters which affect the mass flow rate of refrigerant in the capillary tube can be inlet conditions such as condensing temperature (corresponding inlet pressure), degree of subcooling, and the geometry of the capillary tube such as inner diameter and length. Outlet pressures in real air-conditioners are far lower than the choking pressure, therefore, in general the outlet pressure is not considered as a dominant factor which affect the mass flow rate through the capillary tube.

Figure 7 represents the influence of the degree of subcooling and the length of the capillary tube on the mass flow rate for R22 at the condensing temperature of 45℃. With increased degree of subcooling, mass flow rate increases since the liquid region in the capillary tube is greater with less portion of two phase region where refrigerant has relatively greater specific volume. Capillary tube itself is used as an expansion device, which means that it gives flow resistance to refrigerants. Therefore, the mass flow rate decreased with the length of the capillary tube. Figure 8 shows the influence of the condensing temperature and the inner diameter of the capillary tube on mass flow rate for R22 with degree of subcooling of 1.5℃. The mass flow rate increased with higher condensing temperature since inlet pressure corresponding to the condensing temperature is increased to have higher potential to push refrigerant through a capillary tube. In other words, increased inner diameter of the capillary tube gives wider passage of refrigerant flow, which results in higher mass flow rate if other test conditions are kept the same.

Dimensionless correlation for mass flow rate has been developed in order to predict the refrigerant mass flow rate in an adiabatic capillary tube. Dimensionless parameters are derived by the Buckingham Π theorem (White, 1994; ASHRAE, 1998). Variables which influence mass flow rate are capillary tube geometries, inlet conditions and refrigerant properties. Therefore, the mass flow rate is represented as shown in Eq. (1).

$$\dot{m}=f_1(d,\ L,\ P_{in},\ T_{sub},\ \mu,\ v,\ c_P) \qquad (1)$$

The symbols $\dot{m}$, d, L, P_{in}, T_{sub}, μ, v, and c_P represent mass flow rate, inner diameter, length of the capillary tube, inlet pressure, degree of subcooling, viscosity of refrigerant at the capillary tube inlet, specific volume at the inlet, and specific heat at constant pressure of the refrigerant at

Table 1 Constant and exponents in Eq. (3)

Variable	Value
A	0.1134
a	0.4763
b	-0.3070
c	0.1029

the inlet of capillary tube, respectively. d, μ, v, and c_P are chosen as repeating variables which cannot form a Π group. Therefore, the functional relationship will have the following equivalent form.

$$\frac{\dot{m}}{\mu d}=f_2\left(\frac{P_{in}d^2}{v\mu^2}, \frac{L}{d}, \frac{T_{sub}c_P d^2}{v^2\mu^2}\right) \tag{2}$$

Equation (2) is rewritten in a generic form with unknown exponents as shown in Eq. (3).

$$\frac{\dot{m}}{\mu d}=A\left(\frac{P_{in}d^2}{v\mu^2}\right)^a\left(\frac{L}{d}\right)^b\left(\frac{T_{sub}c_P d^2}{v^2\mu^2}\right)^c \tag{3}$$

Constant A, and exponents a, b, and c are obtained to minimize the deviations between the mass flow rate calculated from Eq. (3) and the experimental results. The constant and exponents are shown in Table 1.

Root mean square (RMS) of the deviation between the calculated values from the correlation shown in Eq. (3) and the experimental results is 7. 9% and the maximum absolute deviation is 20.5% for R22. Figure 9 shows a comparison of the exper-imental data in this study and those of Kuehl and Goldschmidt (1990), Wijaya (1992), Melo et al. (1994), and Chang and Ro (1996b) as a function of mass flow rate calculated by Eq. (3). In this process, thermophysical properties of each refrigerant were put into Eq. (3) to calculate mass flow rate. Kuehl and Goldschmidt (1990) showed experimental data for R22 using a capillary tube with diameter of 1.245 mm and length of 760 mm and found that the root mean square (RMS) of the deviation is 21.3% and the maximum absolute deviation is 28.7%. Wijaya (1992) tested a set of capillary tubes with length of 1524 to 3048 mm for diameters of 0.6604, 0.7874, and 0.8382 mm using R134a and the calculated root mean square (RMS) of the deviation is 8.13% and the maximum absolute deviation is 18.7%. Experimental data reported by Melo et al.(1994) for

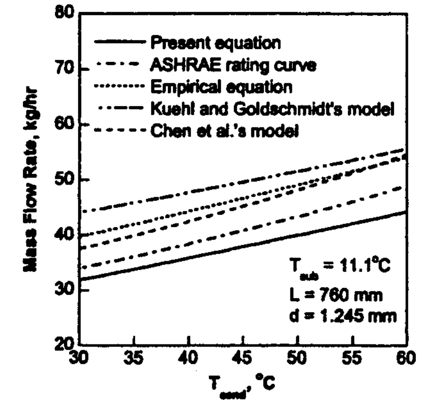


Fig. 10 Comparison of mass flow rates from other sources

R134a using a capillary tube with diameter of 1. 05 mm and length of 2030 mm, and root mean square (RMS) of the deviation is 14.5% and the maximum absolute deviation is 18.6%. Chang and Ro (1996b) presented mass flow rate information for R407C using adiabatic capillary tubes with diameters of 1. 2 and 1.6 mm, and length of 1500 mm and root mean square (RMS) of the deviation is 7.0% and the maximum absolute deviation is 14.4%. As shown in Figure. 9, deviations of measured mass flow rates and calculated mass flow rates were ranging from -18.6 to 28.7%.

In Fig. 10, the critical mass flow rate as a function of condensing temperature for a given capillary tube (d=1.245 mm, L=760 mm) is shown for several models and empirical equations. Values calculated from Eq. (3) are compared with those from the ASHRAE design chart (1994), the empirical equation and the capillary tube model proposed by Kuehl and Goldschmidt (1990, 1991), and the model proposed by Chen et al. (2000). Predictions of critical mass flow rate from the present model are lower than those from other sources. This might have caused due to use of pressure taps, which were used to measure pressure distribution along the capillary tube. As mentioned by Melo et al. (1995), the mass flow rate is reduced by approximately 6% when pressure measurements were taken along the capillary tubes using 9 or 10 pressure taps with a diameter of 0.3 mm made by

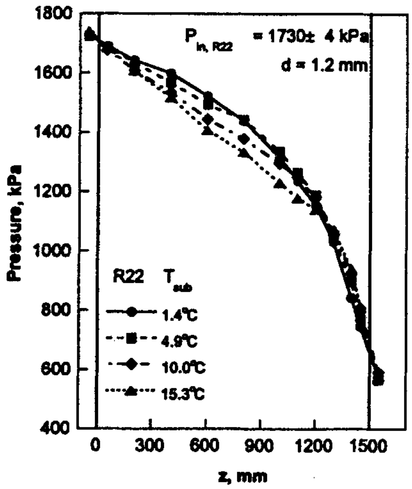


Fig. 11 Pressure drop along the capillary tube for R22 and R407C at the same condensing temperature

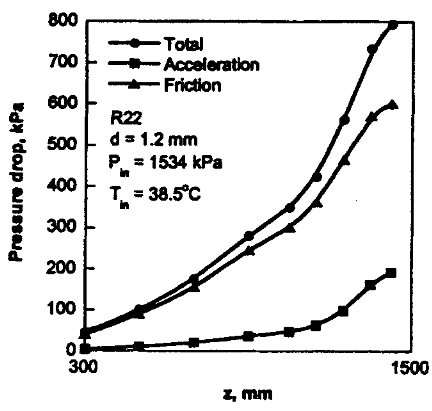


Fig. 12 Pressure drop along the capillary tube to show frictional and accelerational contributions

an electro discharge machine. In our study, 6 to 11 pressure taps with a diameter of 0.1 mm depending on the tube length were made by a laser beam, and it is presumed that the pressure taps raised a disturbance on the flow inside capillary tubes, which may result in a slightly bigger pressure drop and lower mass flow rate. It should be stated that the pressure distribution cannot be measured without pressure taps, and the pressure tap effect should be clarified using a good model for capillary tube flow.

3.3 Pressure distribution along the capillary tube

The pressure distributions along the capillary tube for various degree of subcooling are shown in Fig. 11 for R22 when the average condensing temperature was 40℃ and the capillary tube (length of 1500 mm and inner diameter of 1.2 mm) was used in the test. This figure shows that with increased degree of subcooling the pressure drop in liquid region is greater, which can be conceived by a nearly straight pressure decrease near the inlet of the capillary tube. For higher degree of subcooling, the pressure drop per unit length of the tube is also greater due to increased mass flow rate, and it is also observed that liquid phase region becomes wider. The pressure drop per unit length in liquid phase region due to friction is almost constant because of relatively constant specific volume of refrigerant, while the pressure drop per unit length in two phase region becomes much greater than that in liquid phase region, because the specific volume in two phase region increases drastically as the refrigerant expands.

The pressure drop along the horizontal adiabatic capillary tube is expressed as follows (Collier and Thome, 1996).

$$-\frac{dP}{dz}=f\frac{G^2\bar{v}}{2d}+G^2\frac{d\bar{v}}{dz} \tag{4}$$

The symbols P, z, f, G, $\bar{v}$, and d represent pressure, length, friction factor, mass flux, average specific volume of refrigerant, and inner diameter of capillary tube, respectively. Equation (4) shows that two main causes of the pressure drop in the capillary tube are (1) friction on the wall and (2) acceleration due to the variation of specific volume. Pressure drop due to acceleration is represented as follows (Collier and Thome, 1996).

$$-\left(\frac{dP}{dz}\right)_a=G^2\frac{d}{dz}\left[\frac{x^2V_g}{\alpha}+\frac{(1-x)^2v_f}{(1-\alpha)}\right] \tag{5}$$

$$\alpha=\frac{x}{x+(1-x)\frac{\rho_g}{\rho_f}} \tag{6}$$

In the above equations α, x, and ρ represent void fraction, quality, and density of two phase refrigerant from homogeneous flow model, re-

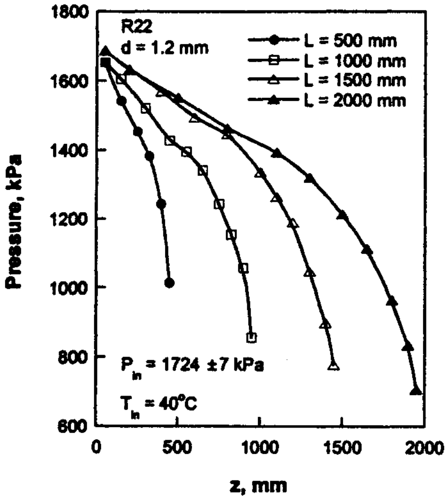


Fig. 13 Pressure drop along the capillary tube for various lengths of capillary tube

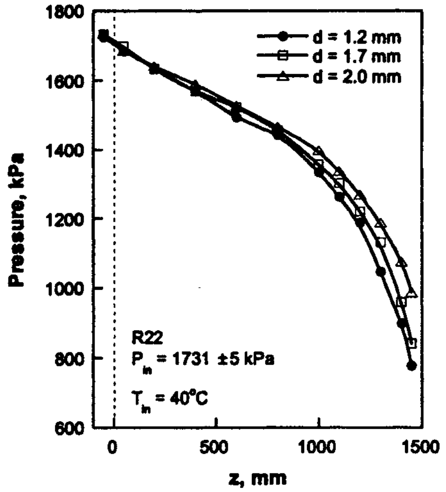


Fig. 14 Pressure drop along the capillary tube for various inner diameters of capillary tube

spectively.

Pressure drop through the two phase region due to friction and acceleration is shown in Fig. 12, which indicates that the main cause of the overall pressure drop is friction throughout the capillary tube, and the pressure drop due to acceleration along the capillary tube increases as the refrigerant flows toward the exit of the capillary tube.

Figure 13 shows a pressure drop with respect to various length of capillary tube. This represents that total pressure drop along the capillary tube increases when the length of the capillary tube increases, but pressure drop per unit length decreases. As shown in Eq. (4), pressure drop per unit length decreases if the mass flow rate is decreased by the increased flow resistance due to increased length of the capillary tube. In addition, Fig. 14 shows pressure drop along the capillary tube of the length of 1.5 m versus the several inner diameters of 1.2, 1.7, and 2.0 mm. As easily expected, the pressure drop is increased when inner diameter of the capillary tube is decreased.

4. Conclusions

The performances of R22 in adiabatic capillary tubes of several length and inner diameter combinations were investigated experimentally. Experiments were conducted in a pump-driven test rig, and the underpressure of vaporization, mass flow rate, and pressure changes were obtained in the test. Underpressure of vaporization depends on the opposite contributions of increased mass flow rate and increased degree of subcooling. It represents the existence of metastable liquid phase in the capillary tubes, and generally it increases with reduced degree of subcooling for the same mass flow rate and with the increased mass flow rate for the same degree of subcooling. In our experiment, increased degree of subcooling resulted in an increase of mass flow rate, therefore, some trend was observed which indicate that the underpressure of vaporization increases with increased degree of subcooling.

The mass flow rate of refrigerants became greater with an increased inner diameter or decreased length of the capillary tube. It also increased when condensing temperature was elevated and the degree of subcooling increased.

The pressure distribution along the capillary tube shows that the pressure drop per unit length increased when inner diameter was reduced and length of the capillary tube shortened. The relative deviation between the calculated values from the correlation of this study and experimental data was 20.5% at its maximum and RMS of deviation was 7.9%. However, deviation between the calculated values and measured values by other researchers ranged from -18.6 to 28.7%.

Acknowledgement

This work has been supported by Samsung Electronics Co., Ltd., Korea Science and Engineering Foundation (Grant No. 1999-1-304-006-3), and BK21 project of the Ministry of Education.

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