International Refrigeration and Air Conditioning Conference
School of Mechanical Engineering

2014

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Puya Javidmand
Wichita State University, United States of America, sxjavidmand@wichita.edu

Masoud Zareh
Department of Mechanical and Aerospace Engineering, Science and Research Branch, mn_zare@yahoo.com

An Experimental Comparison of the Refrigerant Flow along Adiabatic and Diabatic Coiled Capillary Tubes

Masoud Zareh[1], Puya Javidmand[2], Klaus A. Hoffmann[3]

[1]Department of Mechanical and Aerospace Engineering, Science and Research Branch,
Islamic Azad University
Tehran, Iran
m.zareh@srbiau.ac.ir

[2]Department of Aerospace Engineering, Wichita State University,
Wichita, KS
sxjavidmand@wichita.edu

[3]Department of Aerospace Engineering, Wichita State University,
Wichita, KS
Klaus.hoffmann@wichita.edu

ABSTRACT

This paper presents experimental comparison of the R134a flow through lateral coiled capillary tubes-suction line heat exchanger (diabatic) and adiabatic coiled capillary tubes in vapor compression refrigeration cycle (VCRC). The experimental results illustrated that mass flux ratio (G_c/G_s) is main parameter that affects metastable flow through non-adiabatic coiled capillary tube. Therefore, an increase of the mass flux ratio represents a decrease of the heat transfer rate between the capillary tube and suction line. The measured and saturated pressure distribution also proof that metastable flow in the non-adiabatic coiled capillary tube with 1.397 mm inner diameter, 30 mm coil diameter, 4360 mm length, 4 mm inner diameter of suction tube exists, when the heat transfer rate between the coiled capillary tube and the suction line is weak with mass flux ratio more than 343.

1. INTRODUCTION

Capillary tubes are widely used as refrigerant controlling devices, expansion devices and also as heart of a small scale vapor compression refrigeration cycle (VCRC). They connect outlet condenser to the inlet evaporator and balance the refrigeration cycle pressure and control the refrigerant mass flow rate. Whereas capillary tube has no moving parts, they cannot be adjusted to varying load conditions, therefore it was used in small household refrigerators and freezers with nearly constant refrigeration load. In general, the inner diameter and length of a capillary tube ranges are from 0.5-2.0 mm, and 2-6 m, respectively (Zareh et al., 2014). In some vapor compression refrigeration cycle applications, capillary tubes are coiled to minimize the space. The fluid flow in coiled capillary tubes is subjected to the centrifugal force which causes secondary flow effect (Khan et al., 2009, Shokouhmand and Zareh, 2014). Generally, in domestic refrigerators, the capillary tube and suction line consists of an adiabatic section in inlet, a heat exchanger in middle section and an adiabatic section in outlet. They are soldered to each other in order to increase the heat transfer between them, which results in increasing cooling capacity, and also avoid the entering of liquid into the compressor (Domanski et al., 1994). The refrigerant flow through capillary tube flow is significantly affected by the heat transfer to the suction line. The adiabatic flow follows a path that is close to an isenthalpic line, whilst the non-adiabatic flow is projected toward the line of saturated liquid, increasing the amount of liquid in the two-phase mixture and decreasing the vapor quality at the evaporator inlet. As a consequence, the refrigerating effect and also cooling capacity in the evaporator is increased. The efficiency of R134a vapor compression cycle improves notably by suction line heat exchange. Therefore, capillary tube-suction line heat exchangers are used in household refrigerators, while adiabatic capillary tubes are used in room air conditioners. The capillary tube- suction line heat exchanger in a domestic refrigerator such as freezer consists of an adiabatic inlet region, a heat exchanger region and an adiabatic outlet region (Domanski et al., 1994). The refrigerant exiting from the condenser flashes in the adiabatic inlet region and enters the heat exchanger region of the capillary tube, where heat was rejected to the cold suction line downstream of the evaporator and enters the adiabatic outlet region at reduced quality and pressure. Simulation of a non-adiabatic capillary tube is much more difficult rather than of an adiabatic one because of occurrence re-condensation and reverse heat transfer in the non-adiabatic capillary tube. Due to the effect of the metastable flow, the refrigerant mass-flow rate in capillary is affected not only by the working conditions, but also by the way reaching to this condition (Bansal and Yang, 2005). The numerical simulations and experimental validation for R-22 refrigerant flow through coiled capillary tubes were conducted by Zhou and Zhang (2006a, 2010). They also used a homogenous two-phase flow model in the mathematical modeling and three different friction factor correlations. Inlet pressure fluctuations were found during the experiments for coiled adiabatic capillary tubes by Zhou and Zhang (2012). They found that the fluctuation caused due to rapid change of inlet temperature from one point to another point. Their results showed that fluctuation may be occurred by the variation of flash point location in the capillary tube. Moreover, experimental indicated that the pressure fluctuation for coiled capillary tubes is much more prominent than for straight ones; the inlet pressure fluctuation is weak for small coil-diameter capillary tubes.

Garcia-Valladares (2007a, b, c) analyzed different geometries of diabatic capillary tubes–suction line under critical or non-critical pure and refrigerant mixtures flow conditions and steady and unsteady state conditions. He also analyzed re-condensation phenomenon of the refrigerant flow in the heat exchanger region. He also studied steady and unsteady numerical simulation for analyzing of the refrigerant flow characteristics in both adiabatic and diabatic straight capillary tubes (both lateral and concentric configuration) under critical and non-critical refrigerant flow conditions. An one-dimensional mathematical model for simulating of refrigerants flow through adiabatic and non-adiabatic capillary tubes was presented by Hermesa et al. (2001). They proposed and validated their results using a dataset included of more than 1400 experimental data points, for adiabatic and non-adiabatic flow of refrigerants in both lateral and concentric configurations of straight capillary tube–suction line heat exchanger. A numerical simulation by Artificial Neural network (ANN) model for R600a refrigerant flow was presented by Heimel et al. (2013). This model can be used for choked flow, non-choked flow, and also for two-phase inlet conditions. Chingulptak and Wongwises (2010a, b; 2011) presented a numerical method for refrigerant flow through coiled capillary tube using the homogenous model. They found that in the same dimension, mass flow rate through the coiled capillary tube with coil diameter of 40 mm is reduced by about 9% compared to the straight ones.

Xu and Bansal (2002) developed a homogeneous flow model for refrigerant flowing along non-adiabatic straight capillary tube–suction line. Their results indicated that the flow characteristics inside the diabatic capillary tube could be affected by heat transfer and pressure drop of the refrigerant flow. Although it is expected that flashing to the vapor phase starts at the end of single-phase region, it happens with some delay at a point where flow pressure is slightly less than its saturation pressure. This region is called "metastable flow (a thermodynamic non equilibrium condition)" region (Chen and Lin, 2001; and Fiorelli et al., 2013), which is often ignored in numerical simulations. Metastable flow of R-410A through capillary tubes was studied experimentally by Fiorelli et al. (2013). They measured the under pressure of vaporization and metastable region length by using temperature profiles, mass flow rate, inlet pressure, and also friction factor equations for each capillary tube. A correlation based on experimental data with maximum error 22% was developed by them. Existence of the metastable phenomenon was reported by Liu and Bullard (1997) in diabatic straight capillary tube - suction line configuration for sub-cooled refrigerant flow under steady and quasi-steady conditions. Chen and Lin (2001) studied the metastable phenomenon of the R134a flow along a diabatic capillary tube-suction line heat exchanger. They reported a correlation for prediction of under pressure of vaporization based on the classic nucleation theory. Their results indicated that the mass flux ratio plays an important role in the occurrence of metastable flow in this flow condition.

In most of the previous studies reviewed here, straight capillary tubes are invoked in adiabatic or non-adiabatic flow condition for refrigerants R12 and R22. In the other hand, there have been limited numerical or experimental works dedicated to metastable flow phenomenon in coiled capillary tubes particularly in diabatic flow conditions. Therefore, considering the scarcity of devoted literature on metastable and re-condensation flow phenomenon in helically capillary tube. Main objects of this work are comparison, investigation of this phenomenon and determination of the critical mass flux ratio through on-adiabatic flow condition. So, in the present work, an experimental investigation is presented along diabatic coiled capillary tube that predicts the metastable phenomenon for R134a. To this end, an experimental comparison of metastable and re-condensation flow phenomenon is developed for adiabatic and non-adiabatic helical capillary tube and results are verified using previously published relevant data.

2. Experiments and Test Procedure

2.1. Experimental Set Up

The schematic feature of the test set up was shown in Figure 1. This experimental set up included three circuits' refrigerant, glycol ethylene and cooling water. It also can be used for different refrigerants, such as the R134a, under adiabatic and diabatic flow conditions by using two hand valves. This set up was also designed for simulating of refrigerant containing of oil and without oil by using oil separator.

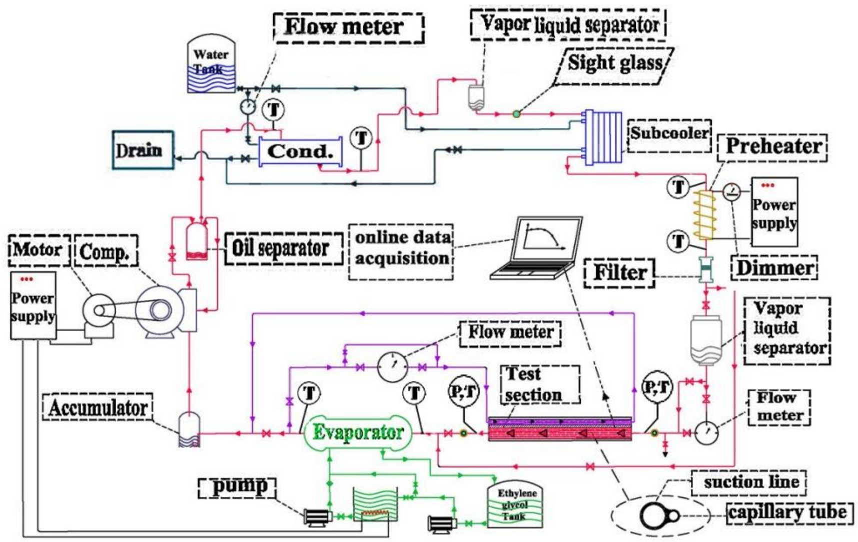

Figure 1: The schematic diagram of experimental set up

Therefore, this experimental set up was adjusted in order to comparison of the adiabatic and non-adiabatic flow conditions. The adiabatic and diabatic test section were made of the copper helical capillary tubes- suction tube that were shown in Figure 2. Diabatic helical capillary tube can be divided into three regions; the adiabatic initial length, the heat exchanger region and the adiabatic final length. In heat exchanger region with lateral arrangement was placed in thermal contact with suction tube with counter refrigerant flow. To avoid unnecessary heat loss to the surroundings of the heat exchanger section, it is wrapped by 40-50mm layer of Styrofoam. The mass flow rate of refrigerant vapor supplied to the suction line was adjusted by hand valve and also was measured by a second counter. Once the desired sub cooling degree of refrigerant was achieved, the refrigerant flow in the test section was adjusted by diverting the excess refrigerant flow into evaporator by bypass circuit and using hand expansion valve. Modular test section was installed between condenser and evaporator. Refrigerant enters evaporator from capillary tube. A tank containing of 100 liters of glycol ethylene and an electric heating coil that was designed for heat load of the evaporator. This system have thermostat in order to adjust the temperature of glycol ethylene that was circulated by a centrifugal pump. Then, the two-phase flow from the evaporator was sent into the accumulator allowing only vapor to enter the reciprocating compressor. One oil separator are installed downstream of the reciprocating compressor in the bypass circuit. The oil-free vapor from oil separator is condensed in the shell and tube condenser. Open loop cooling water was used to obtain steady condition in condenser. In order to visualize the state of refrigerant flow, five sight glasses are installed: first after the condenser, second at the inlet, and third at the outlet of capillary tube and suction line. A compact type sub-cooler and an electrical heater were used after the water cooled condenser to fine tune the sub cooling degree of refrigerant flow. The high pressure liquid refrigerant flow from condenser was collected in the receiver. To continuously supply refrigerant flow through capillary tube, saturated or sub-cooled liquid was collected in the receiver-drier. A filter was used to collect and remove the moisture, impurities, and others particles. A manual needle expansion valve is also placed in parallel with the test section to adjust the excessive refrigerant flow in the capillary tube.

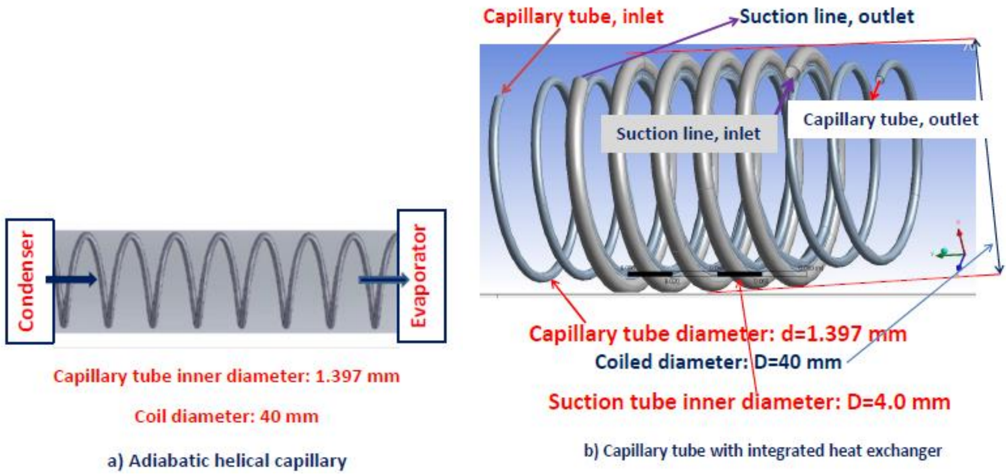

Figure 2: Schematic of employed capillary tubes

2.2. Test Procedure and Conditions

The pressure along the helical capillary tube is measured using 10 online pressure transducers. Temperature along the wall of the capillary tube and suction tube are also measured by 25 K-type thermocouples. Temperature at different locations of the main component such as condenser, sub-cooler and compressor is measured by K-type thermocouples. Moreover, the pressure at the inlet and outlet of the compressor and test section was measured by four bourdon pressure gauges. The helical capillary tubes of two different diameters (0.7874 mm and 1.397 mm), four different lengths (4.36 m-6.2 m), and two coil diameters (30 mm and 40 mm) are used in the experiments. The range of inlet sub-cooling degree is adjusted from 3 °C to 12 °C, while condensing temperature is set at 36 °C to 52 °C for each tube length, tube diameter and coil diameter of the capillary tubes. In order to reach the steady state conditions in each test run, the exit pressure of the capillary tube is adjusted lower than 200 kPa for all test runs that ensures choked conditions. This process takes about 30-60 minutes to accomplish. A steady state condition is achieved when the inlet pressure and condensed temperature remain steady during 5 minutes within ±0.05 bars and ±0.5 °C, respectively. High accuracy counters are used to measure the mass flow rate in the bypass circuit located next to the horizontal test section.

Table 1: Amount of uncertainty for all parameters and measured data

Parameters	Instruments	Uncertainty	Remarks
Capillary tube length	Steel rule	±1.0	
Capillary tube diameter	Optical measuring projector	± 0.025 mm	
Coil diameter	Vernier calipers	±0.1 mm	
Roughness of capillary tube	Surface profilometer	± 0.15 µm	
Mass flow rate	Digital counter	0.11%	
Temperature	Thermocouples (K-Type)	±0.5°C	Online data acquisition system
Pressure	Pressure transducer	0.25%FS	Online data acquisition system, max. ±0.04 bar
	Pressure gauge	5.5 kPa	

The pressure and temperature variation along the coiled capillary tube are recorded using online pressure transducers and online thermocouples with an accuracy ±0.25% of full scale and ±0.5°C, respectively. Finally, the uncertainty of all parameters and measured data was shown in Table 1. Considering the scarcity of data for helical capillary tube in adiabatic and non-adiabatic flow conditions, the present experimental work for straight capillary tubes are compared with the numerical and experimental data reported by Chen and Lin (2001) and also with numerical results of Zareh et al. (2014).

3. Results and Discussions

3.1. Adiabatic helical capillary tubes

Figure 3 illustrates the present experimental work and experimental data of Chen and Lin (2001) for adiabatic straight capillary tubes with 0.66 mm inner diameter, 160 cm length and refrigerant R134a. It is clearly observed that pressure variations is linear in single phase and decreases rapidly in two-phase region near the tube exit due to flashing. Furthermore, this comparison shows a reasonable agreement of the presented numerical results and the measured data. Maximum error of about 5.50% indicates a reasonable agreement of the presented work results and the experimental data of Chen and Lin (2001).

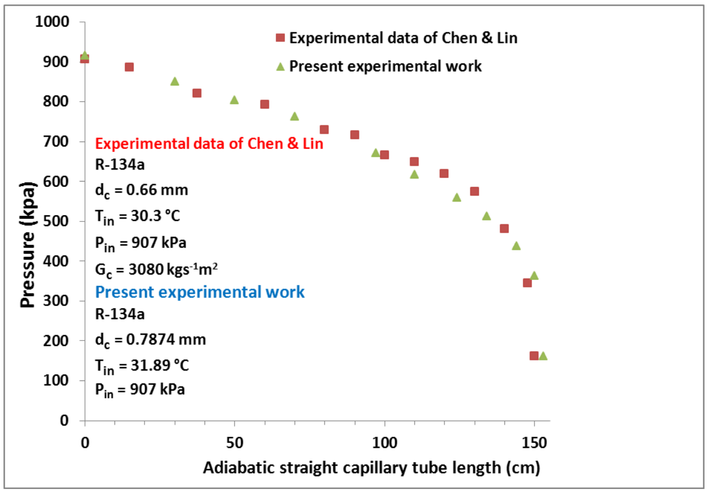

Figure 3: Comparison of the present experimental work and experimental data of Chen and Lin (2001) for R134a flow through adiabatic straight capillary tube

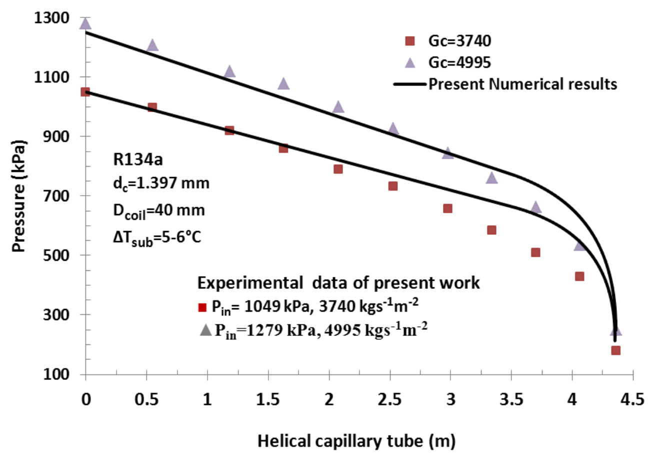

Figure 4: Comparison of present experimental data and numerical results of Zareh et al. (2014) for R134a

Figure 4 compares present experimental pressure profile and results reported by Zareh et al. (2014) through the helical tube for a tube with 1.397 mm inner diameter, 4.36 m length, 40 mm coil diameter, 5-6 °C degree of sub-cooling and R134a for two different inlet pressure and mass flux 3740 and 4995 $kgs^{-1}m^{-2}$. In liquid phase region, pressure gradient is almost constant, but the pressure drops with a sharp gradient close to capillary tube exit due to pressure gradient approaching to infinity, as a result fluid rapidly flashes to vapor phase. It is also observed that there is small deviation and a reasonable agreement between present measured data and numerical results. In addition, there is slight deviation in two-phase region, due to the fact that two-phase metastable flow region was ignored. Moreover, comparison between them displays a maximum error of about 5.5% for pressure distribution.

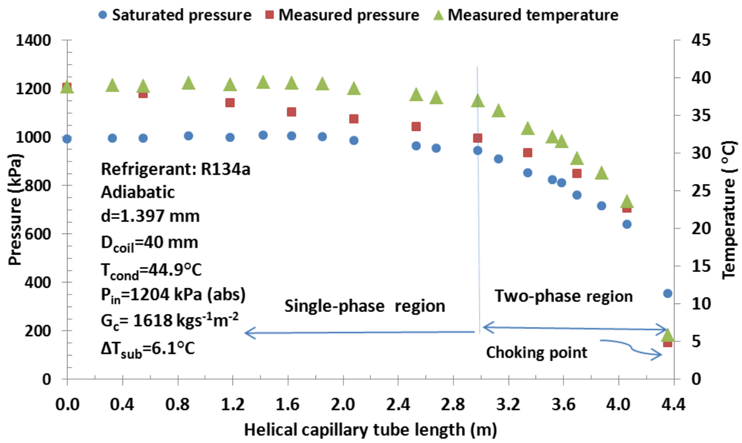

Figure 5: The measured pressure, saturated pressure profiles related to measured temperature of R134a flow through adiabatic helical capillary tube

Figure 5 illustrates distributions of the experimental measured pressure, saturated pressure and measured temperature of refrigerant flow along adiabatic helical adiabatic capillary tube with inner diameter of 1.397 mm. It can be seen that refrigerant flow may be divided into two regions by neglecting metastable region. The saturated pressured and measured temperature remains almost constant before vaporization has occurred at their intersection.

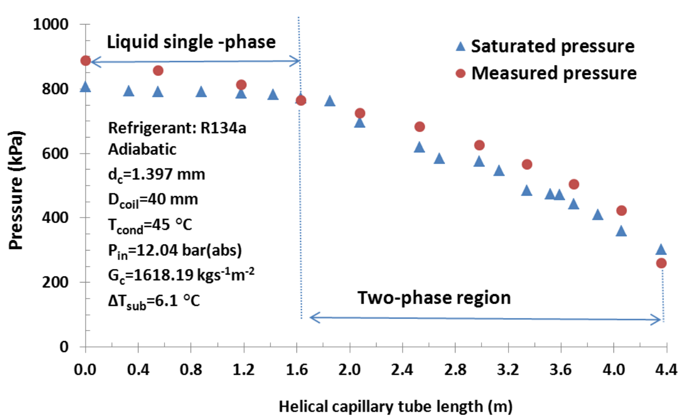

Figure 6: Saturation and measured pressure distribution through the adiabatic capillary tube with inlet temperature of 45°C

Figure 6 displays the pressure and temperature distribution of R-134a through coiled adiabatic capillary tube with inner diameter of 1.397 mm, length of 4.36 and coil diameter of 40 mm. At the first 1.6 m of the tube there is sub-cooled liquid flow and the next 0.4 m length of the tube is the metastable region and through the rest of the tube there is two-phase saturated flow region.

3.2. Diabatic coiled capillary tubes

Figure 7 shows a comparison between the present measured pressure and experimental pressure for non-adiabatic straight capillary tube-suction line for capillary tube 0.66 mm inner diameter, 160 cm length, 100 cm heat exchanger length, 4 mm suction tube, refrigerant R134a and mass flux ratio 251 of Chen and Lin (2001). It is clearly seen that small between pressure profiles and mass flux ratios. Furthermore, this comparison displays maximum error of about 11.5%. All pressure profiles were drawn by considering of ±5 kPa error bars.

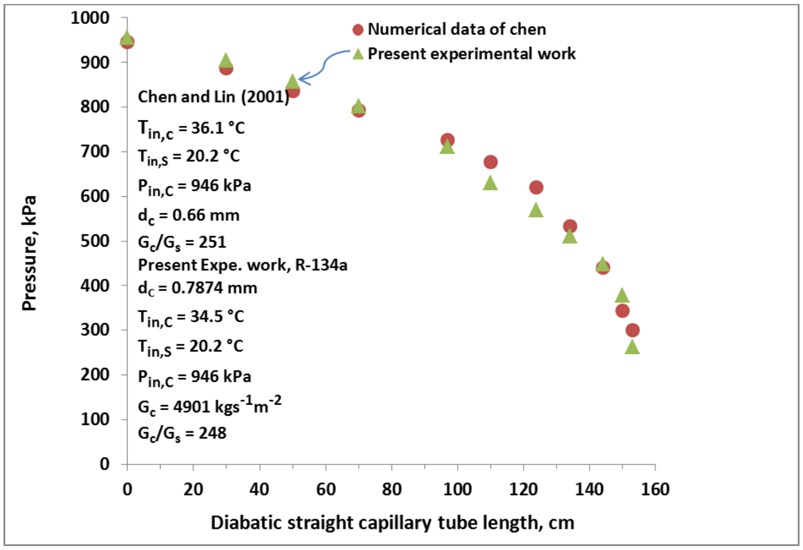

Figure 7: Comparison of the present experimental work and numerical results of Chen and Lin (2001), for R134a flow through non-adiabatic straight capillary tube

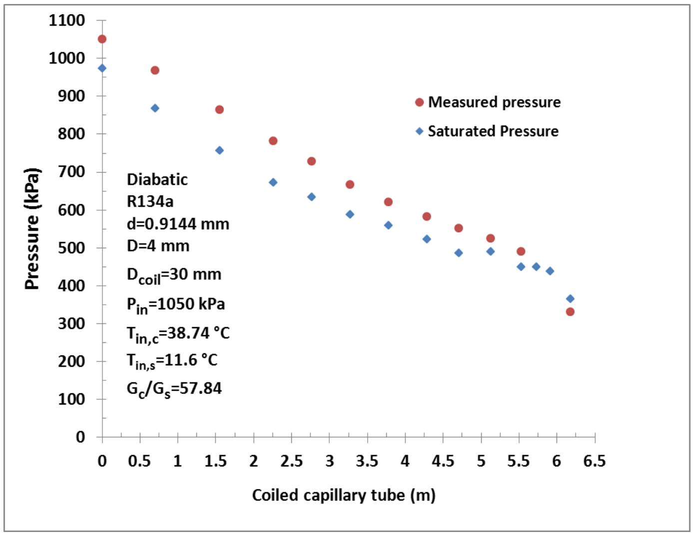

Figure 8: Measured and saturated pressure profile along diabatic coiled capillary tube, without metastable flow

The measured pressure and saturated pressure (related to measured temperature) distribution along diabatic coiled capillary tube-suction line with 0.9144 mm inner diameter, 6.18 m length, 30 mm coil diameter, 4 mm inner diameter of suction tube and refrigerant R134a was shown in Figure 8. In mass flux ratio (G_c/G_s=57.84), that G_c and G_s are defined as refrigerant mass flux through capillary tube and suction line, respectively, the rate of heat transfer between coiled capillary tube and suction line is strong, therefore intersection point measured and saturated pressure profile was occurred near the end of capillary tube. It is clearly observed that the measured pressure drops linear along whole of capillary tube; it means that liquid single phase flow through coiled capillary tube. In other hand, the inception of vaporization is not appeared.

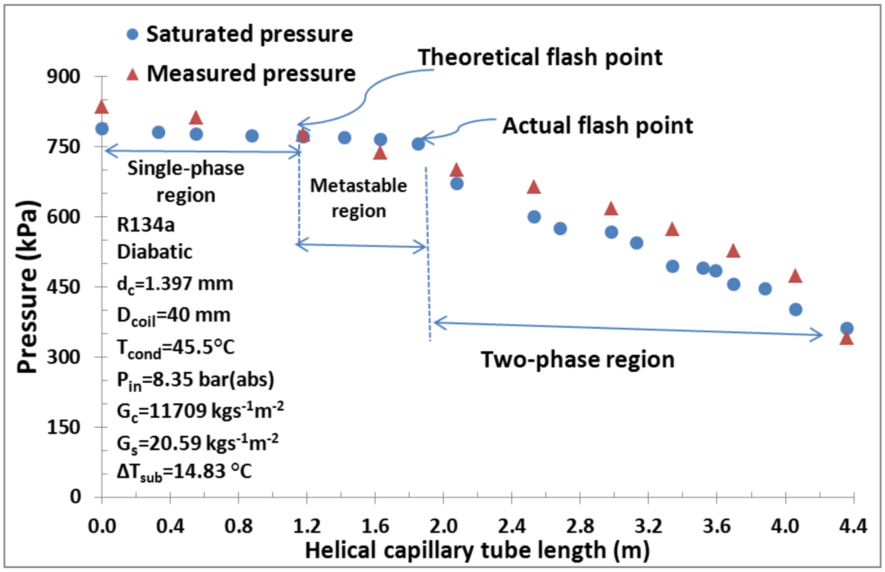

Figure 9: Measured pressure and saturated pressure's profile of R134 flow through non-adiabatic coiled capillary tube

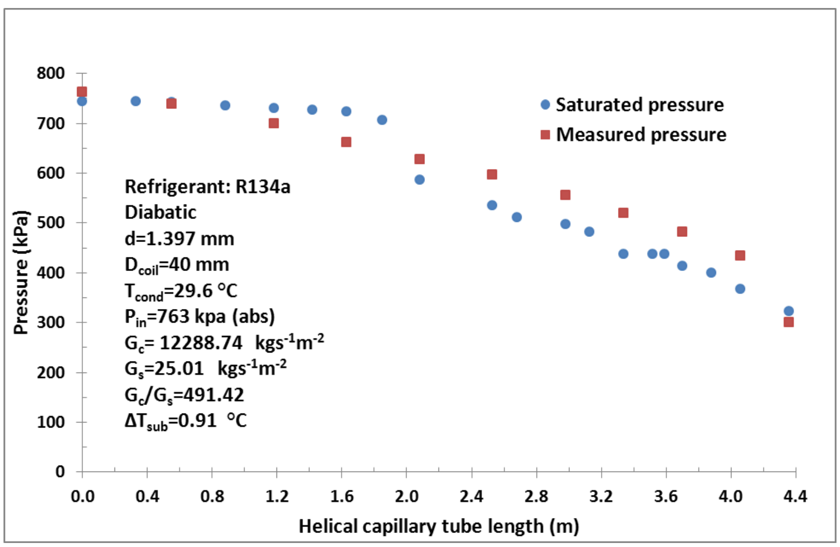

Figure 10: Distribution of measured pressure and saturated pressure along coiled capillary tubes;(metastable flow appears)

Distribution of the measured pressure and saturated pressure through non-adiabatic coiled capillary tube for tube 1.397 mm inner diameter, 4.36 m length, 40 mm coil diameter, 4 mm inner diameter of suction tube and refrigerant R134a was shown in Figure 9. By considering of mass flux ratio, the refrigerant flow inside the capillary tube could be divided three regions; single-phase flow, metastable region and thermodynamic equilibrium two-phase region. Theoretical flash point is intersection measured and saturated pressure profile, but inception of vaporization does not happen (P_S), whereas in actual flash point, vaporization takes place at this location (P_V). These points were occurred about 1.2 m and 2.0 m from inlet of coiled capillary tube. This pressure difference (P_S-P_V) is called "under pressure" that is main characteristic of metastable phenomenon or delay of refrigerant vaporization.

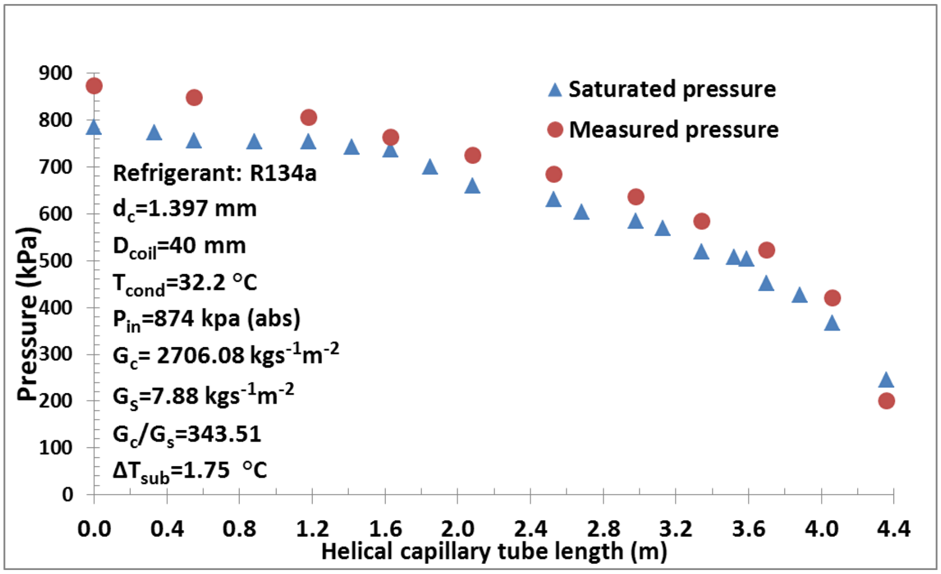

Figure 11: Distribution of measured pressure and saturated pressure along coiled capillary tubes in critical conditions metastable flow (G_c/G_s=343)

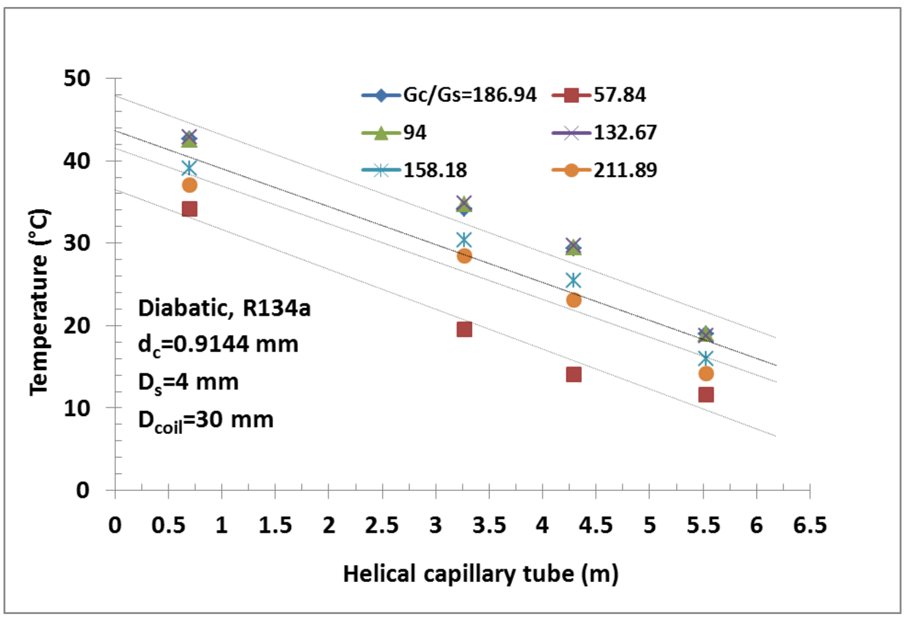

Figure 12: Effects of mass flow rate ratio on distribution of temperature of the refrigerant

Effect of the mass flux ratio was shown in Figure 10 and Figure 11. These Figures illustrate the measured and saturated pressure profile along coiled capillary tube with inner diameter of 1.397 mm, 4.36 m length, coiled diameter 40 mm and refrigerant R134a. In Figure 11, measured pressure profile does not cross saturated pressure profile, but only touch each other in mass flux G_c/G_s=343, that is known critical condition of metastable flow phenomenon, whereas in Figure 10, G_c/G_s=491, by increasing of mass flux ratio, the rate of heat transfer between of capillary tube and suction tube decreased, therefore metastable flow of refrigerant appears in non-adiabatic coiled capillary tubes.

Figure 12 depicts temperature distribution through the suction line for various mass flow rates. It shows that there is linear temperature distribution through the suction tube that is consistent with the presented results by assuming constant convection coefficient of refrigerant flow through the tube.

4. CONCLUSIONS

This paper can be concluded as following:

- Experimental study was presented for R134a flow through both diabatic and adiabatic coiled capillary tubes.
- Comparison between present measured data and experimental data of Chen and Lin (2001) displays a maximum error of 9.5% for pressure distribution.
- The heat transfer rate between capillary tube and suction line decreases by increasing of the mass flux ratio.
- Measured pressure, saturated pressure and temperature variation are obtained experimentally for different mass flux ratio and test condition for both adiabatic and non-adiabatic R134 flow along coiled capillary tube.
- Weak heat transfer rate between the coiled capillary tube and suction line ensure the existence of the metastable phenomena in non-adiabatic flow condition.
- The effect of capillary tube inner diameter, length, and coil diameter, and also various test conditions such as inlet pressure, inlet temperature, and sub-cooling degree of refrigerant was investigated.

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