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Adiabatic Capillary Tube
Test Data for HFC-134a

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Abstract

The function of a capillary tube, one of the most commonly used expansion devices, is to provide a restriction between the high and low side pressures of a refrigeration system. In so doing, it meters refrigerant at the desired flow rate for a given operating condition. At present, no experimental data are available for describing the performance of a capillary tube employing HFC-134a as a working fluid. The purpose of this study is to present experimental test data of adiabatic capillary tube performance for HFC-134a. A set of capillary tubes 5 feet to 10 feet long with inside diameters of 0.026, 0.031 and 0.033 inches was tested. The tests were conducted over a range of condensing temperatures and degrees of subcooling. A modified PAG oil (BRL 150, Allied-Signal Inc. U. S. Patent 4975212) with a viscosity of 150 SUS was used. An oil separator which can be bypassed was installed to determine the effects of oil circulation on the capillary tube performance.

Nomenclature

A	tube cross sectional area	M	refrigerant mass flow rate
D	inside tube diameter	ΔP	pressure difference
ϵ	relative surface roughness	Re	Reynolds number
f	friction factor	V	axial velocity
ΔL	section length	ρ	density

Introduction

HFC-134a, an environmentally safe refrigerant, has been considered as the primary candidate to replace CFC-12, commonly used in refrigeration units. Due to the differences in the thermodynamic and transport properties of HFC-134a from those of CFC-12, a refrigeration unit employing HFC-134a needs to be redesigned to optimize its performance. A new oil also needs to be developed since mineral oil, which is the existing oil used with CFC-12, is not compatible with HFC-134a. The experimental data comparing the adiabatic capillary tube performance for CFC-12 and HFC-134a show that there may not be a direct correlation between the two fluids [1]. Consequently, it is important to have a set of capillary tube performance data for HFC-134a.

The capillary tube, one of the four major components in a refrigeration system, is basically a small bore tube. It was first used in the early 1930's in refrigerators. Today, capillary tubes are used in many types of refrigeration and air-conditioning units. "Capillary" may not be the right name in this context since there is no capillary action inside a capillary tube. However, it is called a capillary tube due to its much smaller inside diameter than those of other tubes used in a refrigeration system. Even though it is mechanically simple, the flow behavior inside a capillary tube is very complex.

Several processes may be used to manufacture a capillary tube: plug-drawn, wire-drawn and sunk tubes. These different processes may result in different surface roughness of a tube. Consequently, there may be different flow and performance results of capillary tubes produced by different manufacturers. As pointed out by Sweedyk [2] in his tests using air flow, two tubes of similar inside diameters with arithmetical average roughnesses of 0.000004 and 0.000016 feet respectively would cause up to 15 % deviation in flow at high inlet pressure. Sweedyk also showed that a refrigeration unit used a capillary tube with a surface roughness of 0.000018 feet and charged with 17.3 oz of HCFC-22 gave a balanced and very acceptable unit. When a second capillary tube with a surface roughness of 0.000006 feet was installed in the same unit using the same refrigerant charge, the refrigerant flooded through the evaporator.

The analysis of the effects of surface roughness on fluid flow in a capillary tube may be simplified and broken down into two categories: the effect of surface roughness on friction factor and the effect of friction factor on fluid flow rate. The analysis of surface roughness may be simplified by studying the case in a liquid flow. The well known correlation for friction factor is developed by Colebrook [3].

$$1/\sqrt{f}=1.14+2*\log(D/\varepsilon)-2*\log\left[1+\frac{9.3}{Re*(\varepsilon/D)*\sqrt{f}}\right] \qquad (Eq.1)$$

Eq. 1 is implicit in terms of f and iterations are required to solve for f. Figure 1 shows the friction factor as a function of relative surface roughness for different tube inside diameters. At two mass flow rates, the relative surface roughnesses chosen for the analysis with Eq. 1 are 0.000004 and 0.000016 feet respectively. The values of transport and thermodynamic properties of HFC-134a for the calculations are taken at 110°F. For the mass flow rate of 0.25 lb/min, the friction factor is increased by about 28 % when the relative surface roughness is increased from 0.000004 feet to 0.000016 feet. However, the difference in friction factor is increased further by about 35 % for the higher mass flow rate.

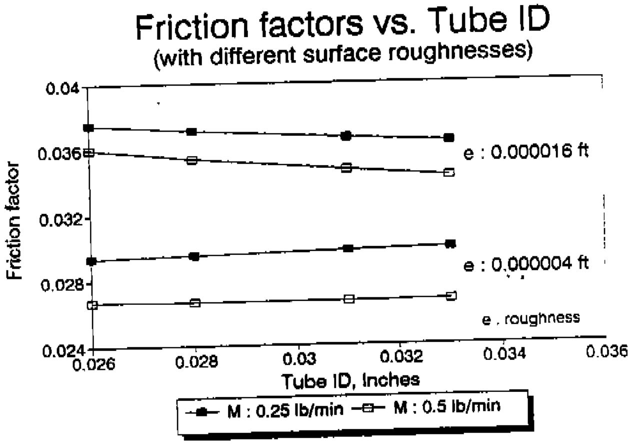


Figure 1. Effect of surface roughness on friction factor

Similarly, the effect of friction factor on the refrigerant mass flow rate in a tube can be analyzed by using a simplified case, liquid flow. Pressure drop in the liquid region may be calculated as follow: [4]

$$\Delta P = \frac{f*\Delta L*\rho*V^2}{2*D} \qquad (Eq.2)$$

The mass equation for steady flow :

$$M = V\,A\,\rho \qquad (Eq.3)$$

By substituting Eq. 3 into Eq. 2 and rearranging the result using $A = (\pi D^2/4)$:

$$\Delta P = \frac{f\,\Delta L\,M^2}{\rho\;2\;(\pi/4)^2\;D^5} \qquad (Eq.4)$$

Solving Eq. 4 for mass flow rate, M:

$$M = \sqrt{\frac{\Delta P\;\rho\;(\pi/4)^2\;2\;D^5}{f\;\Delta L}} \qquad (Eq.5)$$

It is clear from Eq. 5 that an increase in friction factor by 28% or 35% will lead to a decrease in mass flow by 12% and 14% respectively assuming other things are held constant. It is very important to note that this simplified analysis only considers flow in liquid region. In reality, the flow in a capillary tube consists of single and two phase flows. However, the analysis done above shows that the surface roughness may affect the mass flow in a capillary tube.

In the present study, the surface roughness of the capillary tubes used in the experiments was not the main focus. Consequently, in order to generate consistent experimental data, the capillary tubes used were all obtained from the same manufacturer.

For certain refrigerants, it is a common practice to attach the capillary tube to the suction line. This is done to reduce the refrigerant quality at the tube exit. As a result, the efficiency of a refrigeration system will be increased. It can be shown thermodynamically that for CFC-12 and HFC-134a the system efficiency of a refrigeration unit may be improved if the capillary tube is attached to the suction line. However, the adiabatic capillary tube test data for HFC-134a is an important first step to provide some insights and point of reference in analyzing the non-adiabatic capillary tube performance. In this study, correlations of adiabatic capillary tube performance for HFC-134a are presented. The analysis of non-adiabatic capillary tube performance for HFC-134a will follow upon the completion of this study.

Test Facility

The test facility is shown in Figure 2. It is a standard vapor compression cycle with the additions of a subcooler, a superheater and an oil separator.

Condenser/evaporator

The condenser is a coaxial heat exchanger with refrigerant flowing in the inner tube and water flowing in the annulus. The inlet water temperature is controlled by an electric heater and the flow rate by a needle valve. For the evaporator, an electric heater with an adjustable power input is used as the heat source.

Subcooler/superheater
A subcooler and a superheater are used to fine tune the refrigerant inlet condition at the capillary tube. The subcooler is a coaxial heat exchanger with water flowing in the outer tube and refrigerant flowing in the inner tube. The water circuit is connected to a refrigerated circulator bath. An electric heater with an adjustable power input is used as the superheater.

Compressor
An hermetic rotary type compressor is used to circulate refrigerant in the system. An oil separator is installed at the compressor outlet and can be bypassed when it is not needed.

Expansion device
Two types of expansion devices were installed: a hand expansion valve and the capillary tube being tested. The hand expansion valve is used during the start-up to keep the suction line at a positive pressure, thus avoiding any possibility of air leakage into the system. The capillary tube is heavily insulated to prevent any heat loss or heat gain. The capillary tube connections are shown in Figure 3.

Instrumentation
Type T thermocouples are used to measure temperatures at various locations. Two pressure transducers are used to measure the capillary tube inlet and exit pressures A Coriolis type mass flow meter is used to measure the refrigerant mass flow rate through the capillary tube being tested. All the output signals from these transducers are recorded through a computerized data acquisition system.

Table 1. Instrument specifications

Item	Range	Accuracy
Thermocouple	up to 662 °F	+/- 0.4 °F
Pressure transducer	0-150 Psia : 0-500 Psia	+/- 0.1% full scale
Flow meter	0-2 lb/min	+/- 0.24%

All thermocouples were calibrated by immersing them in a refrigerated bath. A precalibrated RTD and water-ice mixture were used as the reference points. The mass flow meter accuracy was checked by allowing water to flow within and weighing the amount of the water after a certain time interval. The pressure transducers were calibrated by using a factory precalibrated dial pressure gage.

Special treatments of deburring and washing were employed for each capillary tube prior to any testing. Deburring is an operation performed on a capillary tube to remove the slight amount of metal which is disturbed by the sawing operation and which tends to restrict the entrance and outlet of the tube. Washing is done by using a solvent (CFC-11) to remove grease and dirt inside a capillary tube.

During data taking, the high side pressure vs. time was plotted on the monitor screen to see if the system had reached a steady state. It usually took a minimum of one hour for the system to reach a steady state. The inlet temperature and pressure to the capillary tube were maintained steady within +/- 0.4 °F and +/- 1.0 psia from the set points respectively. The outlet pressure at the capillary tube was uncontrolled since the discharge was sonic. Once a steady state was obtained, the data were taken over a one minute period and were then averaged.

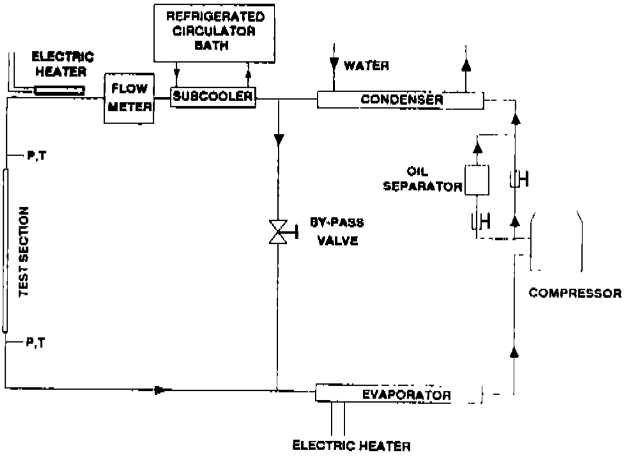
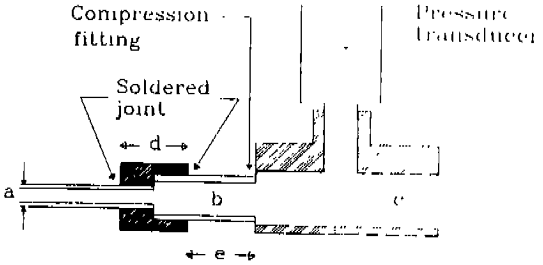


Figure 2. Schematic diagram of the test rig



a: capillary tube outside diameter
b: 0.075 inches inside diameter
c: 0.090 inches inside diameter
d: 0.25 inches length
e: 0.8 inches length

Figure 3. Capillary tube coupler

Effect of oil circulation

Figures 4 and 5 show the experimental data of capillary tubes of 0.026 and 0.033 inches inside diameters run with and without the oil separator. It was observed that the data obtained by using an oil separator did not differ significantly from those obtained without an oil separator.

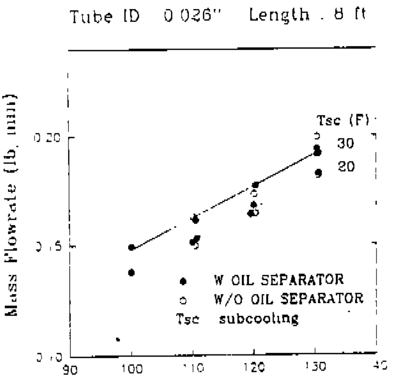


Figure 4. Effect of oil circulation

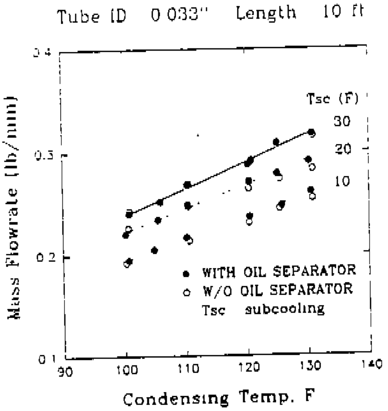


Figure 5. Effect of oil circulation

Results and Discussions

Capillary tubes of 0.026, 0.031 and 0.033 inches inside diameters were tested. A range of condensing temperatures from 100° F to 130° F with 10° F to 30° F of subcooling was chosen for the operating conditions. The exit pressure at the capillary tubes was uncontrolled since the discharge was sonic. The range of the capillary tube lengths tested was 5 ft to 10 ft. However, for the 0.026 inches inside diameter, tube lengths were limited to 9 ft and the degrees of subcooling were limited from 20° F to 30° F. The reason was that beyond these ranges the capillary tube exit pressure was below atmospheric pressure.

Figure 6 shows the typical plot of temperature distribution along an adiabatic capillary tube. The flow inside a capillary tube may be broken into two regions: liquid and two-phase flows. For a given inlet subcooling, the refrigerant experiences pressure drops as it flows. In this region the temperature of refrigerant remains unchanged as depicted by smooth and straight lines. As the pressure drop increases, part of the refrigerant flashes into vapor. The point where refrigerant flow becomes two-phase flow is called the flash point. The flash point is not the saturation temperature at the operating pressure since metastable or superheated liquid region does exist in a capillary tube [5]. Beyond this point the temperature of refrigerant is dropping as depicted by a sudden change in the temperature distribution plot. The higher the degree of subcooling the later the flash point is. The delay in flash point, as a result of increased subcooling, causes more refrigerant to be passed in a capillary tube since liquid flow offers less resistance than two-phase flow.

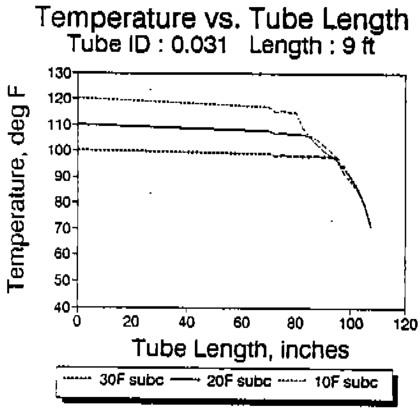


Figure 6. Temperature distribution plot

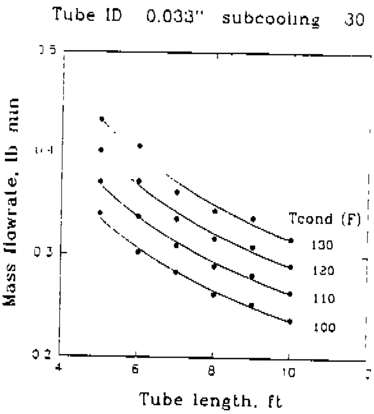


Figure 7. Flow rate vs. tube length

Figures 7, 8, and 9 show a typical family of plots for HFC-134a mass flow rate vs. capillary tube length at different condensing temperatures and degrees of subcooling under adiabatic conditions. Some of the data shown in these figures are averaged values.

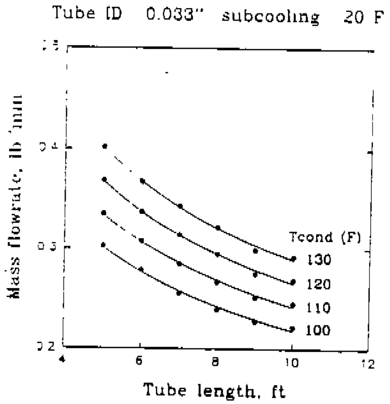


Figure 8. Flow rate vs. tube length

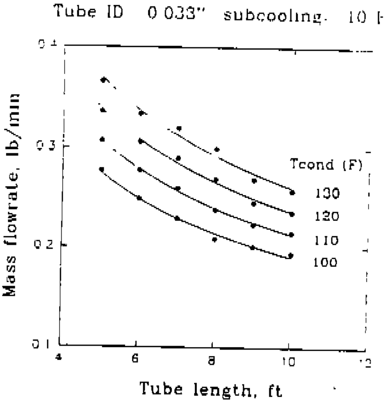


Figure 9. Flow rate vs. tube length

The experimental results were curve fitted and shown in Tables 2, 3 and 4. The regression analysis shows that the best curve fit for the experimental data is in the form of $M=aL^b$ where M and L are the refrigerant mass flow rate in lb/min and capillary tube length in feet respectively. Characters a and b are constants for each curve fit.

Table 2. Adiabatic performance for capillary tube of 0.033" inside diameter

SC (°F)	ID (inches)	L (ft)	T cond (°F)	a	b	CC
10	0.033	5-10	100	0.6486	-0.5326	0.9942
			110	0.7107	-0.5217	0.9985
			120	0.7724	-0.5126	0.9955
			130	0.8349	-0.5053	0.9886
20	0.033	5-10	100	0.6382	-0.4661	0.9953
			110	0.7134	-0.4699	0.9973
			120	0.7879	-0.4725	0.9977
			130	0.8623	-0.4746	0.9973
30	0.033	5-10	100	0.7481	-0.4983	0.9964
			110	0.7994	-0.4821	0.9963
			120	0.8500	-0.4679	0.9932
			130	0.9020	-0.4563	0.9882

Table 3. Adiabatic performance for capillary tube of 0.031" inside diameter

SC (°F)	ID (inches)	L (ft)	T cond (°F)	a	b	CC
10	0.031	5-10	100	0.6216	-0.5888	0.9811
			110	0.6737	-0.5792	0.9905
			120	0.7271	-0.5721	0.9950
			130	0.7800	-0.5657	0.9960
20	0.031	5-10	100	0.6468	-0.5474	0.9833
			110	0.6947	-0.5359	0.9894
			120	0.7431	-0.5265	0.9933
			130	0.7903	-0.5176	0.9953
30	0.031	5-10	100	0.6870	-0.5407	0.9936
			110	0.7214	-0.5161	0.9943
			120	0.7572	-0.4956	0.9941
			130	0.7928	-0.4777	0.9934

where SC : degrees of subcooling
ID : tube inside diameters
CC : correlation coefficient
L : tube length
T cond : condensing temperature

Table 4. Adiabatic performance for capillary tube of 0.026" inside diameter

SC (°F)	ID (inches)	L (ft)	T cond (°F)	a	b	CC
20	0.026	5-9	100	0.3055	-0.3940	0.9748
			110	0.3601	-0.4204	0.9923
			120	0.4162	-0.4427	0.9929
			130	0.4721	-0.4602	0.9869
30	0.026	5-9	100	0.3749	-0.4615	0.9763
			110	0.4203	-0.4651	0.9930
			120	0.4655	-0.4676	0.9911
			130	0.5109	-0.4702	0.9805

Conclusions

The adiabatic performance for capillary tubes of 0.026, 0.031 and 0.033 inches inside diameter with HFC-134a are presented. The presence of an oil separator did not have any significant effects on the performance of a capillary tube employing HFC-134a used with BRL 150 oil.

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