1986

Two-Phase Flow in a Diabatic Capillary Tube

M. B. Pate

D. R. Tree

TWO-PHASE FLOW IN A DIABATIC CAPILLARY TUBE

MICHAEL B. PATE
Department of Mechanical Engineering
Iowa State University, Ames, Iowa (USA)

DAVID R. TREE
Department of Mechanical Engineering
Ray W. Herrick Laboratories, West Lafayette, Indiana (USA)

1. INTRODUCTION

Many refrigeration cycles operate using a capillary tube that transfers heat to the suction line through a solder connection. This is often called a capillary tube-suction line heat exchanger. The small diameter and long length of the capillary tube ensures that sufficient restriction exists between the high pressure condenser and the low pressure evaporator to control the refrigerant flow through the system. Heat is transferred from the capillary tube to the suction line because the refrigerant entering the capillary tube is significantly warmer than the refrigerant entering the suction line. The state of the refrigerant flow in the suction line is superheated vapor, while the capillary tube flow is significantly more complicated. The flow regions in the capillary tube starting at the inlet are adiabatic subcooled flow, subcooled flow with heat transfer, two-phase flow with heat transfer, and, finally, adiabatic two-phase flow. In addition, choked flow may exist at the exit point.

The two-phase flow region in a capillary tube that has heat transfer to a suction line has received very little attention in the past. Rather, most capillary tube studies have focused on adiabatic two-phase flow. The differences between these two cases are considerable. For example, in the diabatic capillary tube, heat is transferred from the refrigerant to the tube wall, while the refrigerant simultaneously flashes because of frictional effects. The phenomenon of a flashing flow that has heat transfer to a wall is unusual and quite unlike any of the classical two-phase flow modes such as boiling, condensation, or two-component flow. Another complicating factor is the small diameter of the capillary tube, which increases the relative roughness of the tube so that a smooth assumption is no longer valid.

There are no reports in the literature of experiments that resulted in the determination of the state path in the two-phase flow region of a diabatic capillary tube. In contrast, adiabatic capillary tubes have been studied thoroughly as shown in a recent survey article on the subject /1/. Even though past studies have not measured exact state paths, several experimental capillary tube-suction line heat exchanger studies have been reported in the literature. Some of these are investigations by Swart /2/, Bolstad and Jordan /3/, Christensen and Jorgensen /4/, and Pate and Tree /5/. Swart measured the pressure along a capillary tube that was attached to a suction line, but other data were not reported. In contrast, Bolstad and Jordan measured the wall temperature along the heat exchanger, but capillary tube pressures and suction line fluid temperatures were not reported. Christensen and Jorgensen varied the heat exchange position, but their measurements were limited to inlet and outlet conditions only. Pate and Tree instrumented the length of a capillary tube-suction line heat exchanger so that the capillary tube pressure, wall temperature, and suction line fluid temperature could be measured. These data have been reported for different inlet and outlet conditions /5/.

In this study, the thermodynamic state in the two-phase flow region of a capillary tube was determined using energy balances along the length of the heat exchanger and experimental pressure and temperature data from Pate and Tree /5/. Observations of these state paths and a comparison with an adiabatic capillary tube flow provide significant insight into the two-phase flow region of a capillary tube-suction line heat exchanger. Plots of quality, void fractions, entropy, enthalpy, energy components, momentum components, and two-phase friction factors are presented in this study. Additional insights into the two-phase flow are achieved by plotting flow paths on several widely used two-phase flow pattern maps.

2. EXPERIMENTAL APPARATUS

The experimental apparatus, which used refrigerant R-12 as the test fluid, has been described in the literature previously /5,6/. In summary, the setup consisted of a high pressure accumulator/heater assembly located at the inlet of the capillary tube and a downstream reservoir maintained in a saturated condition using a chilled water system. This setup allowed complete control of boundary conditions, such as the inlet pressure and temperature and the downstream exit pressure, to the capillary tube. Low temperature air was supplied to the suction line to simulate a superheated refrigerant returning to the compressor. Air was a satisfactory substitute in terms of simulating an appropriate convection heat transfer coefficient and heat capacity rate /6/. It also had the advantage of minimizing fluid leakage in the region where thermocouple leads were inserted through the tube wall.

The dimensions of the capillary tube were 2953 mm by 0.71 mm ID. The suction line had an inside diameter of 6.35 mm and was soldered to 2086 mm of the capillary tube. This resulted in the first 702 mm and the last 165 mm of the capillary tube being unattached, thus forming adiabatic flow regions.

The capillary tube was instrumented for fluid pressure and wall temperature, while the suction line was instrumented for fluid temperature. These measurements were made using pressure transducers and thermocouples installed at 305 mm intervals along the heat exchanger. Experimental uncertainties were approximated as ±0.2°C for temperature, ±1 kPa for pressure, and ±0.1 kg/hr for mass flow rate.

3. EXPERIMENTAL DATA

Even though some experimental data have been reported previously /5/, a thorough analysis of these data in the two-phase flow region using actual state paths has not been performed. Figure 1 is a plot of saturation temperatures that correspond to pressure measurements along the capillary tube. The different curves represent a range of inlet subcoolings along with a representative adiabatic test case.

Figure 1 shows that the flash point is delayed as the inlet subcooling is increased as evidenced by a decrease in the length of the nonlinear temperature profile region. Since a two-phase flow results in a greater flow restriction than single-phase flow, the mass flow rate through the capillary tube increases. The adiabatic test case is shown for comparison purposes. It not only has a lower flow rate, but the pressure gradient is greater in the two-phase region. Both phenomena can be explained by noting that the quality is much higher in the adiabatic capillary tube, since heat transfer from the capillary tube suppresses the formation of vapor and, hence, lowers the quality. This effect will be discussed in greater detail when the data are analyzed in a later section.

Figure 2 is a plot of temperatures for the capillary tube fluid, tube wall, and suction line fluid along the capillary tube-suction line heat exchanger. The plot is for a typical test case and illustrates the type of data available for analysis in this study. Specifically, these data along with energy balances were used to solve for quality along the heat exchanger. This quality and saturation pressure data were then used to define the thermodynamic state in the two-phase flow region of the capillary tube.

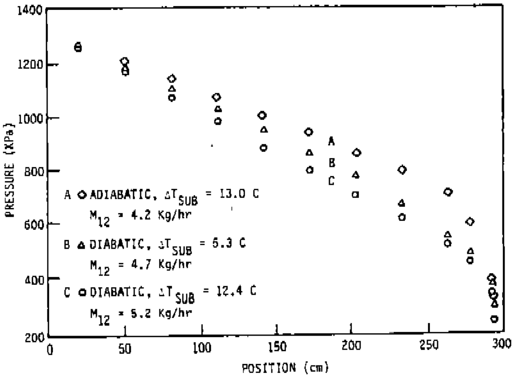

Fig. 1 - Capillary tube pressure.

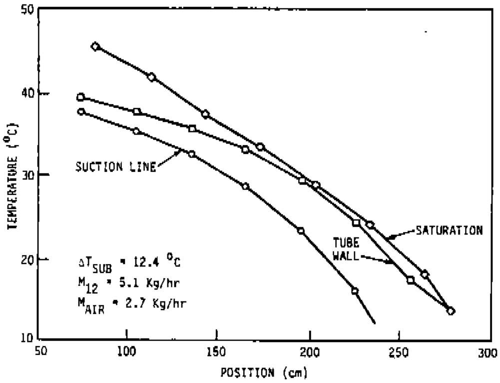

Fig. 2 - Diabatic capillary tube temperature.

Figure 2 also illustrates the difficulty in finding the point of saturation (i.e., flash point) that separates the subcooled region from the two-phase flow region. This difficulty results from the data being measured at 305-mm intervals along the tube and from the film temperature drop on the capillary tube side being quite small and unknown. The procedure for locating the flash point required analyzing plots similar to Fig. 2. For each test case, the two-phase region was assumed to start at the pressure measurement point closest to the region where the capillary tube fluid and wall temperatures converged. This method resulted in a flash point uncertainty of approximately ± 153 mm. Past investigators of adiabatic capillary tubes did not experience a problem in locating the flash point because the large increase in quality just downstream of the flash point resulted in a well-marked, nonlinear pressure profile. Additionally, in subcooled adiabatic flow, the wall temperature is constant and equal to the fluid temperature so that the flash point is well marked at the point where the saturation temperature curve intersects the constant wall temperature line.

4. QUALITY AND VOID FRACTION

Since the capillary tube pressure was measured in the two-phase flow region along the heat exchanger, only one additional property was required to determine the thermodynamic state. Quality was selected as the second property since it can be calculated from experimental data and energy balances along the heat exchanger. Specifically, energy balances can be written on incremental lengths of the capillary tube so that the energy leaving the capillary tube refrigerant is equal to the energy transferred to the suction line and surroundings. Homogeneous and equilibrium two-phase flow was assumed so that the resulting energy balance equation for an increment of length, 1 to 2, along with supporting equations is as follows:

$$0 = \dot{m}_c\ (\bar{h}_2 - \bar{h}_1) + \dot{m}_c\ G_c^2\ \bar{v}_{avg}\ (\bar{v}_2 - \bar{v}_1)$$

$$+ \dot{m}_s\ c_{p_s}\ (T_{s_2} - T_{s_1}) + U(z_2 - z_1)(T_w - T_{amb}) \tag{1}$$

where

$$\bar{h} = h_f(1 - x) + h_g x \tag{2}$$

$$\bar{v} = v_f(1 - x) + v_g x \tag{3}$$

$$\bar{v}_{avg} = \frac{\bar{v}_1 + \bar{v}_2}{2} \tag{4}$$

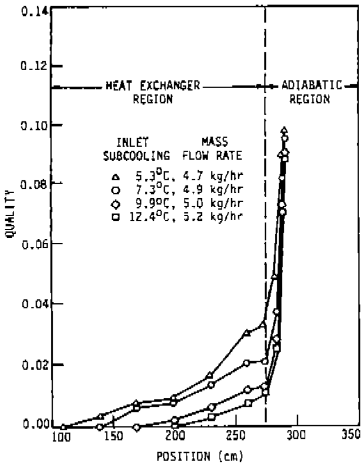

Fig. 3 - Quality in the capillary tube.

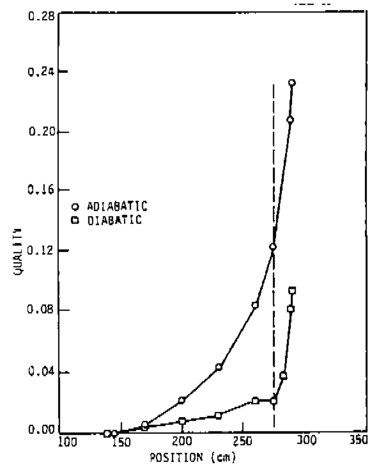

Fig. 4 - Quality comparison for adiabatic and diabatic capillary tubes.

$$T_w = \frac{T_{w_2} - T_{w_1}}{2} \tag{5}$$

The quality at position 2 is the only unknown variable in the above equations, assuming that position 1 corresponds to either the flash point or else to a position where the thermodynamic state is known. Therefore, the thermodynamic state at a given location, position 2, can be specified since two properties, quality and saturation pressure, are known.

Equilibrium qualities calculated from eqs. (1) through (5) have been plotted in Fig. 3 as a function of position for several diabatic test cases. The data shown are for inlet subcooled conditions ranging from 5.3 to 12.4° C with flash point positions ranging from 108 cm to 200 cm from the capillary tube inlet. Interestingly, the slopes of the quality profiles are approximately linear or else only slightly curved in the heat exchanger region. It also appears that the increase in quality is considerably restrained in the heat exchanger region as compared to the adiabatic flow region near the exit; specifically, the quality gradient for the adiabatic region is at least a factor of ten greater than the gradient for the heat exchanger region. Since most of the quality change occurs in the adiabatic region, the actual exit qualities are quite close, in the range of 0.09 to 0.10. This occurs even though the flash point positions are quite different.

Figure 4 compares a diabatic and an adiabatic test case when the flash points occur at similar positions, approximately 140 cm from the capillary tube inlet. As noted previously, heat transfer from the capillary tube acts to restrain the increase in quality. At a position that corresponds to the exit of the heat exchanger, the quality is a factor of six higher for the fully adiabatic capillary tube. Additionally, the increase in quality is approximately linear for the heat exchanger case and obviously nonlinear for the adiabatic case, as was noted previously.

Void fraction profiles provide additional insight into the two-phase flow region of a capillary tube-suction line heat exchanger. For homogeneous two-phase flow, void fractions are defined as the ratio of the vapor volume to the total volume occupied by the vapor and liquid. Void fractions can be calculated from quality as follows:

$$\alpha = \frac{x\,v_g}{(1 - x)v_f + xv_g} \tag{6}$$

It should be noted that even small mass fractions of vapor (low qualities) can occupy large fractions of the volume, especially at the lower pressures that exist near the exit. Figure 5 shows void fractions along the capillary tube for two different inlet

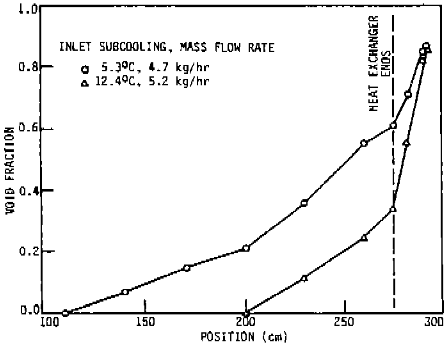

Fig. 5 - Void fraction in the capillary tube.

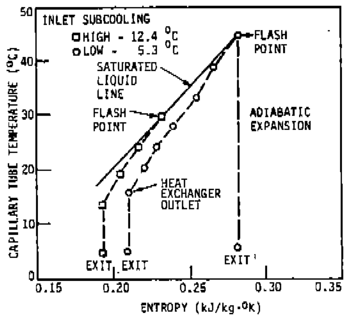

Fig. 6 - Temperature-entropy diagram and two-phase expansion path.

subcoolings. Interestingly, the volume fractions are almost equal at the exit point. This can be explained by the fact that the qualities are quite close, within ten percent of each other, and the exit pressures are slightly different.

5. THERMODYNAMIC STATE PATHS

Significant insight into the two-phase flow region of a capillary tube-suction line heat exchanger can be gained by plotting the actual thermodynamic state path on refrigerant R-12 property diagrams. In the past, capillary tube-suction line heat exchanger processes have been shown as approximations on property diagrams since exact paths were unknown. The qualities calculated in the previous section along with the pressure data can now be used to determine the thermodynamic state paths in the two-phase flow region of the capillary tube. Some of the property diagrams presented in this section are a temperature-entropy (ts) diagram, an enthalpy-entropy (hs) diagram, and a pressure-enthalpy (ph) diagram. These represent some of the more common property diagrams used for analyzing refrigeration systems.

Figure 6 shows the state path for the two-phase flow region plotted on a temperature-entropy (ts) diagram for two different conditions of inlet subcooling. As expected, the test case with the higher inlet subcooling reaches a saturated condition at a lower temperature, resulting in a shorter two-phase flow region. These tests show that the entropy in the capillary tube decreases, which is the result of heat being removed. In contrast, the entropy increases in the adiabatic region downstream of the heat exchanger near the outlet. The entropy also increases over the whole tube length for the adiabatic test case. These increases in the adiabatic region are difficult to observe in Fig. 6 because of the entropy scale. Figure 6 also reinforces a previous observation that the quality in diabatic flow is much less than the quality in adiabatic flow, since the diabatic cases are much closer to the saturated liquid line.

An enthalpy-entropy (hs) property diagram along with a state path for a capillary tube is shown in Fig. 7. Because of the low quality in the capillary tube, the state path deviates only slightly from the saturated liquid line. The decrease in enthalpy and entropy over the length of the heat exchanger occurs because heat is transferred from the capillary tube to the suction line. The adiabatic region at the exit of the capillary tube-suction line heat exchanger has been magnified in Fig. 8. Even though heat is not being removed, the enthalpy of the adiabatic capillary tube decreases slightly. The reason for this decrease is that the kinetic energy is increasing along the tube because of the formation of the vapor phase. Figure 8 also shows the entropy increasing as a result of the unrestrained expansion of the fluid near the exit. An area that is not being investigated herein is two-phase choked flow, which may occur at the exit of the capillary tube when the increase in entropy reaches a maximum. An analysis of choked flow in capillary tube-suction line heat exchangers can be found in /7/.

A pressure-enthalpy (ph) property diagram is shown in Fig. 9. This type of diagram is especially useful for refrigeration system analysis because the change in enthalpy in the evaporator indicates the amount of heat removed by the evaporator. This is commonly referred to as the refrigerating effect /8/. Since the outlet of the capillary tube is also the inlet to the evaporator, the refrigerating effect is increased as the capillary tube transfers heat to the suction line as shown in Fig. 9.

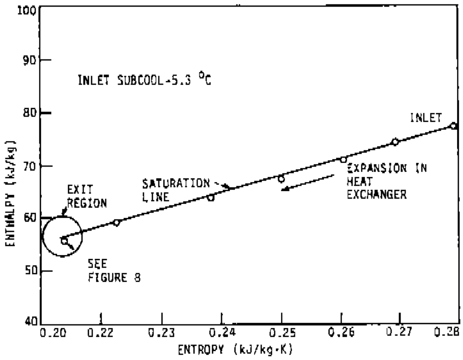

Fig. 7 - Enthalpy-entropy diagram and two-phase expansion path.

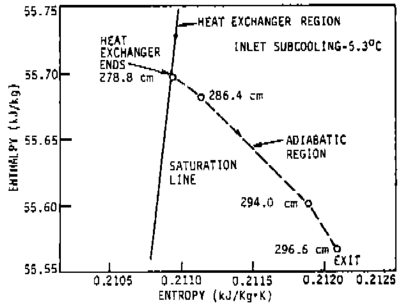

Fig. 8 - Enthalpy-entropy diagram and two-phase adiabatic path.

In addition, the refrigerating effect increases with inlet subcooling. It should also be noted that the mass flow rate through the system increases with inlet subcooling. Therefore, the energy that can potentially be removed by the evaporator increases with inlet subcooling because of both increased enthalpy changes and increased mass flow rates. Figure 9 also shows that minimizing the adiabatic region at the outlet, by increasing the length of the diabatic region, can result in a decrease in outlet enthalpy and, thus, an increase in the refrigerating effect.

6. MOMENTUM BALANCE

A momentum balance performed over an increment of length using experimental data can provide additional insight into diabatic capillary tube flow. The momentum balance consists of a total pressure change, a pressure change due to friction, and a pressure change due to acceleration as follows:

$$\text{Total:} \qquad \Delta P_t = P_2 - P_1 \tag{7}$$

$$\text{Friction:} \qquad \Delta P_f = \frac{G^2\, \bar{v}_{avg}\, f_{2\phi}(z_2 - z_1)}{2D_c} \tag{8}$$

$$\text{Acceleration:} \qquad \Delta P_a = G^2(\bar{v}_2 - \bar{v}_1) \tag{9}$$

A momentum balance results in

$$\Delta P_t = \Delta P_f + \Delta P_a \tag{10}$$

Since pressures and temperatures were experimentally measured along the capillary tube and qualities were calculated from the energy equation, all of the variables in eq. (10) are known except for the two-phase friction factor, $f_{2\phi}$. Therefore, pressure drop components can be calculated and analyzed along the capillary tube length. Friction factors are reported and discussed in the next section.

Components of the pressure gradient along the capillary tube are plotted in Fig. 10 for a typical experiment. This figure shows that the acceleration pressure drop is quite small in the heat exchanger region, less than two to four percent of the total pressure drop. Therefore, the total pressure drop gradient in the two-phase region is approximately equal to the frictional pressure drop gradient as follows:

$$\frac{P_2 - P_1}{z_2 - z_1} \approx \frac{G^2\, \bar{v}_{avg}\, f_{2\phi}}{2D_c} \tag{11}$$

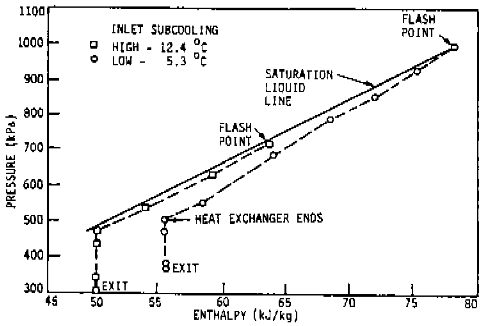

Fig. 9 - Pressure-enthalpy diagram and two- phase expansion path.

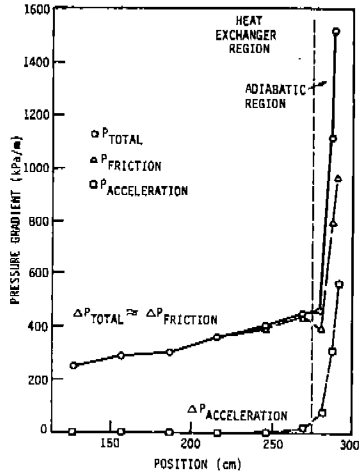

Fig. 10 - Components of momentum balance.

It should be noted that even though this approximation is similar to the linear relationship for incompressible flow, it is actually quite nonlinear because $\bar{v}_{ave}$ and $f_{2\phi}$ change along the capillary tube length as the quality increases. Figure 10 also shows that acceleration changes are quite important for adiabatic flow as evidenced by the large acceleration change at the outlet of the capillary tube-suction line heat exchanger.

7. TWO-PHASE FRICTION FACTOR

Friction factor correlations for single-phase and two-phase flow are usually derived experimentally. Additionally, most friction factor studies for two-phase flow have been for large, smooth tubes that are limited to either adiabatic flow or flow with boiling heat transfer. Since none of these conditions exist in the capillary tube-suction line heat exchanger, an analysis of the experimental data is necessary before a two-phase friction factor correlation can be defined. The all-liquid friction factor can be calculated from the following experimentally derived equation:

$$f = \frac{3.49}{R_e^{0.47}} \tag{12}$$

$$f = 3.49 \left[\frac{\pi D_c \mu}{4\dot{m}_c} \right]^{0.47} \tag{13}$$

It should be noted that this equation is unique for the capillary tube tested in this study; therefore it may not be applicable to other capillary tubes. In addition, it may not resemble other equations in the literature because capillary tubes are quite small (0.71 mm ID) and the walls are rough compared to the small diameter of the tube.

Several methods of calculating a two-phase friction factor have been proposed in the literature. One would expect these methods to show the two-phase friction factor decreasing as the quality, and hence flashing, increases. The reason for this decrease is that the refrigerant viscosity decreases because of the lower viscosity of the added gas phase. The effect of viscosity on Reynolds number and hence friction factor can also be observed in eq. (13). Three of the most successful methods show this trend, as they are based on using a two-phase viscosity in the single-phase friction factor equation. These methods are

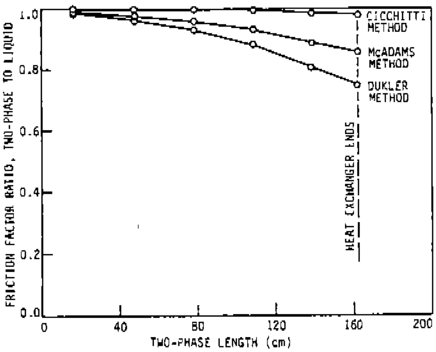

Fig. 11 - Two-phase-to-liquid friction factor ratio.

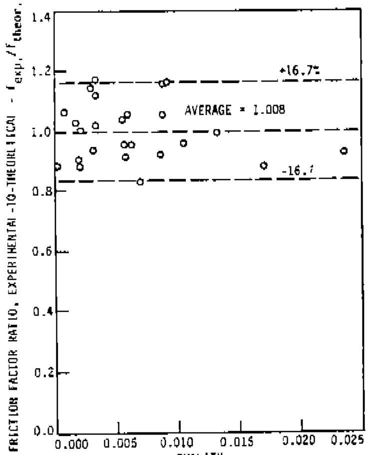

Fig. 12 - Experimental-to-theoretical friction factor ratio for the Dukler method.

McAdams et al. /9/:

$$\frac{1}{\bar{\mu}} = \frac{x}{\mu_g} + \frac{(1 - x)}{\mu_f} \tag{14}$$

Cicchitti et al. /10/:

$$\bar{\mu} = x\mu_g + (1 - x)\mu_f \tag{15}$$

and Dukler et al. /11/:

$$\bar{\mu} = \frac{(xv_g\mu_g + (1 - x)v_f\mu_f)}{\bar{v}} \tag{16}$$

Two-phase friction factors, which have been predicted using the above equations, have been plotted in Fig. 11 as a ratio of the all-liquid friction factor along the length of the capillary tube for a typical experimental test case. The Dukler Method results in the largest decrease in the friction factor because of flashing. Since local two-phase friction factors were calculated in the previous section as part of an investigation on the momentum balance, the most applicable method for determining two-phase friction factors was obtained by plotting experimental-to-theoretical friction factor ratios for a wide range of data. Using this approach, the most accurate method will result in a ratio equal to unity. The Dukler Method appeared to be the most accurate method; the results are shown in Fig. 12. Experimental data scatter in Fig. 12 is caused by uncertainties in the flash point location and pressure measurements. The uncertainty of the pressure measurement is ± 1.0 kPa, which corresponds to a pressure drop uncertainty of ± 2.0 kPa. This uncertainty is ± 16.7 percent of a typical 12.0 kPa measured pressure drop. Even though the Dukler Method appeared to be the most accurate, the McAdams Method was also quite accurate.

A potential limitation exists for using two-phase viscosities to calculate two-phase friction factors. This limitation is that the single-phase friction factor equation, eq. (13), was derived from experimental data measured at a maximum Reynolds number of 12,370, while the two-phase Reynolds numbers were as high as 20,000. A possible reduction in accuracy may occur at the higher two-phase Reynolds numbers.

8. FLOW PATTERN MAPS

The adiabatic capillary tube research reported in the literature has generally assumed homogeneous flow. This assumption is based on observations of bubble or mist flow in small diameter glass tubes /12,13,14/. These glass tube studies, however, may not be applicable to capillary tube-suction line heat exchangers since they cannot simulate the high heat fluxes and the rough wall conditions that exist in actual capillary tubes. Therefore, the homogeneous flow assumption was investigated by plotting experimental data on several well-established flow pattern maps. Three flow pattern maps were selected for this analysis:

- Baker Map /15/
- Hewitt and Robert Map /16/
- Modified Baker Map for Homogeneous and Nonhomogeneous Transition /17/

It should be noted that even though flow pattern maps are generally based on a narrow set of two-phase flow conditions, their application to a variety of other two-phase flows is widely accepted. Even so, their accuracy can be questionable.

The Baker Map /15/ is one of the most widely used two-phase flow pattern maps in the literature. Even though it is based on a narrow range of tube diameters from one to four inches, investigators such as Soliman /18/ have used it successfully with smaller diameter tubes and refrigerant R-12. Experimental data measured along the length of the capillary tube are plotted on a Baker Map in Fig. 13. The flow pattern is shown to be bubbly flow, which agrees with the homogeneous flow assumption.

The Hewitt-Roberts Map /16/ uses the momentum fluxes of the liquid and gas phases for its coordinate axes. Experimental data plotted in Fig. 14 reaffirms the homogeneous assumption with the exception that some separated flow may possibly be occurring near the exit.

A modified Baker plot that has been reduced to only two regions, homogeneous and nonhomogeneous, was proposed by Choe et al. /17/. This map is based on a wide range of data from a variety of sources that include many refrigerant type flows. The consistency of the data signifies a universal transition between homogeneous and nonhomogeneous type flows. According to Fig. 15, the capillary tube-suction line

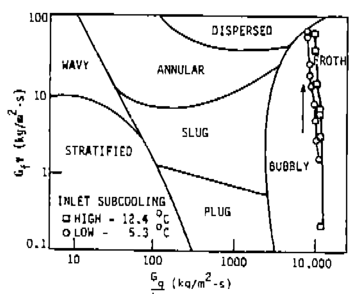

Fig. 13 - Baker flow pattern map.

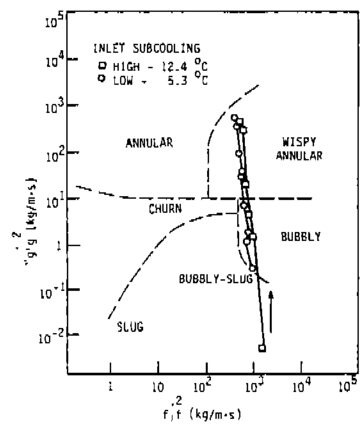

Fig. 14 - Hewitts and Roberts flow pattern map.

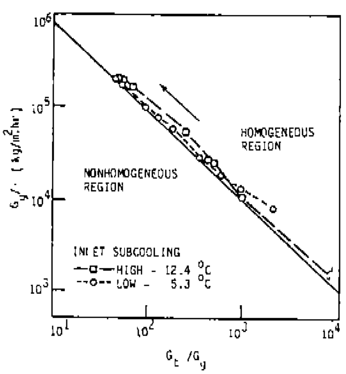

Fig. 15 - Modified Baker map for homogeneous and non-homogeneous transition.

flow is in fact homogeneous. Choe et al. also concluded that any flow greater than 2710 kg/s·m^2 could be considered homogeneous. Most of the experimental tests reported herein resulted in mass fluxes ranging from 3300 to 3650 kg/s·m^2. It is interesting to note that the same capillary tube, operated adiabatically, had mass fluxes from 2250 to 2950 kg/s·m^2, which may or may not be homogeneous.

9. SUMMARY AND CONCLUSIONS

The thermodynamic state in the two-phase flow region of a capillary tube-suction line heat exchanger was determined using energy balances and experimental data. The data measured were capillary tube pressure, tube wall temperature, and suction line fluid temperature.

The increase in quality along diabatic capillary tubes is approximately linear. This is in contrast to the quality changes in adiabatic capillary tubes where rapid increases are evident. A pressure-enthalpy plot showed that the refrigerating effect of an evaporator can be increased by maximizing the diabatic region and minimizing the adiabatic region of a capillary tube. A plot of momentum components showed that acceleration effects are negligible in diabatic capillary tubes. The Dukler Method was shown to be the best method for predicting two-phase friction factors. Finally, using several different flow pattern maps, we verified that the flow in diabatic capillary tubes was homogeneous flow.

NOMENCLATURE

- C_p specific heat, J/kg·°K
- D tube diameter, mm
- f friction factor
- G mass velocity, kg/s·m^2
- h enthalpy, J/kg
- $\bar{h}$ mixture enthalpy, J/kg
- $\dot{m}$ mass flow rate, kg/hr
- P pressure, Pa
- U heat transfer conductance, W/°K·m
- v specific volume, kg/m^3
- $\bar{v}$ mixture specific volume, kg/m^3
- x quality
- z position, cm or m
- α void fraction
- μ viscosity, N·s/m^2
- $\bar{\mu}$ two-phase viscosity, N·s/m^2

Subscripts

- a acceleration
- amb ambient
- avg average
- c capillary tube
- f liquid
- f frictional
- g vapor
- s suction line
- t total
- w wall
- 1,2 positions at each end of tube increment
- 2ϕ two-phase

ACKNOWLEDGMENTS

This research was financially supported by the Whirlpool Corporation. Their support and helpful suggestions are greatly appreciated by the authors.

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Summary

Many refrigeration cycles operate using a capillary tube that transfers heat to the suction line through a solder connection. This is often called a capillary tube-suction line heat exchanger. The flow regions in the capillary tube starting at the inlet are adiabatic subcooled, diabatic subcooled, diabatic two-phase, and adiabatic two-phase. The diabatic two-phase region is the most complicated of the flow regions in that flashing occurs in conjunction with heat transfer to a wall. Therefore, it is quite unlike any of the classical two-phase flow modes such as boiling, condensation, or two-component flow. In this study, the thermodynamic state in the two-phase flow region of a capillary tube was determined using energy balances along the length of the heat exchanger and experimental pressure and temperature data.

The results of the study showed that the increase in quality along diabatic capillary tubes is approximately linear. This is in contrast to quality changes in adiabatic capillary tubes where rapid increases are evident. A pressure-enthalpy plot showed that the refrigerating effect of an evaporator can be increased by maximizing the diabatic region and minimizing the adiabatic region of a capillary tube. A plot of momentum components showed that acceleration effects are negligible in diabatic capillary tubes. The Dukler Method was shown to be the best method for predicting two-phase friction factors. Finally, using several different flow pattern maps, the flow in diabatic capillary tubes was verified as being homogeneous flow.

RÉSUMÉ

De nombreux cycles de réfrigérations utilisent un tube capillaire qui transmet la chaleur vers le niveau d'absorbtion, par l'intermédiaire d'une soudure de liaison. En général, on appelle cela un échangeur de chaleur à tube capillaire et à niveau d'abosrbtion. Les régions d'écoulement dans le tube capillaire, qui partent de l'emtrée, sont sous-refroidies adiabatiques, sous-refroidies diabatiques, diabatiques et adiabatiques ā deux phases. La region diabatique ā double phase est la plus compliquée des régions d'écoulement, c'est dans cette région-là qu'a lieu simultanément une projection avec transfert de chaleur vers la paroi. Cela diffère donc completement de n'importe quel autre modele d'écoulement classique à deux phases, tel que l'ébullition, la condensation et l'écoulement ā deux constituants. Dans cette étude, on a déterminé l'état thermodynamique de la région d'écoulement ā deux phases d'un tube capillaire, en se servant d'équilibres en énergie sur toute la longueur de l'échangeur de chaleur, de la pression expérimentale et de la température.

Les résultats de cette étude ont montré que l'amélioration de la qualité pour ce qui concerne les tubes capillaires diabatiques est à peu prēs linéaire. Cela contraste avec les changements de qualité des tubes capillaires adiabatiques, pour lesquels des améliorations rapides peuvent être constatées. Une partie de la pression-enthalgie a montré que le refroidissement d'un évaporateur peut être augmenté en donnant son maximum ā la région diabatique et en affaiblissant la région adiabatique du tube capillaire. Une partie des composantes de la force a montré que les effets d'acceleration sont négligeables dans les tubes capillaires diabatiques. La méthode de Dukles a été présenteé comme la meilleure méthode pour prévoir les facteurs de friction en deux phases. Finalement, en utilisant diverses méthodes d' écoulement, l'écoulement dans les tubes capillaires s'est révélé homogēne.