Eckels, S.J. and M.B. Pate, *Evaporation and condensation of HFC-134a in a smooth tube and a micro-fin tube.* ASHRAE Transactions, 1991. **97**(2): p. 68-78.

EVAPORATION AND CONDENSATION OF HFC-134a AND CFC-12 IN A SMOOTH TUBE AND A MICRO-FIN TUBE (RP-630)

S.J. Eckels*
Student Member ASHRAE

M.B. Pate, Ph.D., P.E.
Member ASHRAE

ABSTRACT

Evaporation and condensation heat transfer coefficients were measured for smooth and micro-fin tubes with HFC-134a and CFC-12. Micro-fin tubes are internally enhanced tubes that are characterized by numerous small fins that spiral down the tube. For example, in this study, the micro-fin tube had 60 fins with a height of 0.2 mm and an 17° spiral angle. Heat transfer measurements were performed on 3.67 m (12 ft) long tubes with inside diameters of 8.0 mm (0.31 in). Test conditions varied from 5°C to 15°C for evaporation and 30°C to 50°C for condensation. The refrigerant mass flux was varied from 130 kg/m²·s (95,860 lb/ft²·h) to 400 kg/m²·s (294,000 lb/ft²·h).

When HFC-134a was compared to CFC-12 at similar mass fluxes in smooth tubes, the evaporation and condensation heat transfer coefficients were about 40% and 25% higher, respectively. A more relevant comparison of heat transfer coefficients is at equivalent cooling (or heating) capacities. In this case, the HFC-134a heat transfer coefficients were about 10% higher than CFC-12 values for both evaporation and condensation. The micro-fin tube produced higher heat transfer coefficients and pressure drops for all conditions when compared to the smooth tube. For example, for HFC-134a, heat transfer enhancement factors (defined as the convective heat transfer coefficients for the micro-fin tube divided by the value for the smooth tube measured at similar conditions) varied from 1.5 to 2.5 during evaporation and from 1.8 to 2.5 during condensation. Pressure drop penalty factors

(defined similarly to enhancement factors)for both refrigerants were usually less than the heat transfer enhancement factors. However, in the case of HFC-134a at the lowest temperature and highest mass flux, the penalty factor slightly exceeded the enhancement factor.

INTRODUCTION

This paper presents the results of an experimental study to evaluate the performance of HFC-134a during evaporation and condensation inside micro-fin tubes. These results are related to two important issues facing refrigerating and air-conditioning industries: the phaseout of CFC-12 for a suitable non-CFC alternative, such as HFC-134a, and the use of enhanced tubing in evaporators and condensers. The first issue has important implications on the problem of ozone layer depletion, while the second issue is important in terms of increasing cycle efficiency.

Micro-fin tubes, such as the one evaluated herein, are characterized by numerous small fins that spiral down the inside surface of a tube, usually made of copper. Micro-fin tubes are available commercially in geometries that have 50 to 70 fins per tube, fin heights of 0.10 mm to 0.20 mm (0.004 in to 0.008 in), and spiral angles of 8° to 30°. The fin tip and valley shape can vary with the manufacturer being either flat, sharp, and rounded. Most of the micro-fin tubing available commercially is made from copper because manufacturing processes have been successfully

S.J. Eckels is Research Assistant and M.B. Pate is Professor, Department of Mechanical Engineering, Iowa State University, Ames, Iowa

developed to make the miniature fins in copper tubing. Also, nearly all of the applications for micro-fin tubing to date have been in the refrigerating and air-conditioning industries, which utilize copper tubing in evaporators and condensers on a large scale.

MICRO-FIN TUBE LITERATURE REVIEW

Heat transfer and pressure drop in micro-fin tubes have been studied extensively for Refrigerants 22 and 113 and to a lesser degree for Refrigerant 12. Even though it is not used as a working fluid, the results of past studies with CFC-113 are important in understanding fundamental phenomena. CFC-113 was a popular choice in these past fundamental studies because its relatively low vapor pressure makes it ideally suited for laboratory research use.

An extensive literature search on micro-fin tubes and other forms of in-tube enhancement was reported by Schlager et al. (1987). They concluded that of all the forms of in-tube enhancement studied, the micro-fin tube showed the largest increases in heat transfer enhancement relative to the increase in pressure drop enhancement (referred to as pressure drop penalty). This relative increase in heat transfer compared to pressure drop for the micro-fin tube is the primary reason that micro-fin tubes are being utilized in refrigerating and air-conditioning applications. For evaporation, a survey of six publications by Schlager et al., which represented a large range of micro-fin tube geometries, showed typical values of heat transfer enhancement factors varying from 1.3 to 2.0, while pressure drop penalty factors were significantly less. For condensation, a survey of three publications showed typical values of heat transfer enhancement factors varying from 1.6 to 2.0. Again, pressure drop increases were described as either slight or very little.

Since Schlager et al. (1987) published their survey of micro-fin tube literature, a number of additional studies on micro-fin tubes have been published. Specifically, from 1987 to the present, 11 additional papers are found in the literature. These publications are listed, along with the type of refrigerant and the objectives of each study, in Table 1. The majority of these studies are for R-22 during both evaporation and condensation. A detailed literature review is presented in the next section.

Khanpara et al. (1987a) studied micro-fin tubes during evaporation of CFC-113 in both electrically heated tubes (3.8 m long) and fluid-heated tubes (1.0 m long). They observed that for electrically heated tubes, which provided local heat transfer coefficients, the heat transfer enhancement factor for the micro-fin tube varied from 1.18 to 2.72, depending on heat flux and mass flux. In contrast, the maximum increase in pressure drop was about 30%, which corresponds to a pressure drop penalty factor of 1.3. The heat transfer coefficients for the micro-fin tube were comparable for fluid and electrical heating. However, for a smooth tube tested under similar conditions, differences of as much as 20% to 50%, with the electrical heating values being higher, were observed.

Khanpara et al. (1987b) compared the evaporation performance of an electrically heated micro-fin tube operating with two different refrigerants, CFC-113 and HCFC-22. This comparison was important because laboratory tests are often performed with CFC-113, while many air-conditioning systems operate with HCFC-22. For both test fluids, heat transfer enhancement factors decreased with increasing mass flux. In addition, the enhancement factors for HCFC-22 (ranging from 1.2 to 1.7) were less than the enhancement factors for CFC-113 (ranging from 1.3 to 2.2).

Table 1

Eckels, S.J. and M.B. Pate, *Evaporation and condensation of HFC-134a in a smooth tube and a micro-fin tube.* ASHRAE Transactions, 1991. **97**(2): p. 68-78.

Micro-Fin Tube Studies Published Since The Schlager et al. (1987) Survey

Author(s) and Date	Evaporation, Condensation or both	Refrigerant	Description
Schlager et al. (1987)	both	all	survey of in-tube augmentation techniques
Khanpara et al. (1987a)	evaporation	R-113	comparison of heat transfer coefficients for electrical and fluid heating
Khanpara et al. (1987b)	evaporation	R-22, R-113	comparison of heat transfer coefficients for two refrigerants
Reid et al. (1987)	evaporation	R-113	comparison of several augmentation techniques including the micro-fin tube
Schlager et al. (1989a)	Both	R-22	comparison of three different 9.5 mm OD micro-fin tube geometries
Schlager et al. (1989b)	both	R-22	comparison of three different 12.7 mm OD micro-fin tube geometries
Schlager et al. (1988,1989c)	both	R-22	micro-fin tube performance for refrigerant-oil mixtures
Schlager et al. (1989d)	both	R-22	study of oil viscosity effects on micro-fin tubes
Schlager et al. (1989e)	both	R-22	comparison of a micro-fin tube and a low-fin tube
Schlager et al. (1990a)	both	R-22	literature search on oil effects in refrigerant flow in tubes including micro-fin tubes
Schlager et al. (1990b	both	R-22	design equations to predict the performance of micro-fin tubes

Reid et al. (1987) used CFC-113 to compare the evaporation performance of an electrically heated micro-fin tube to the performance of several other types of enhanced tubes. The heat transfer enhancement factors for the micro-fin tube varied from 1.3 to 1.7, while comparable values were also observed for a low-fin tube (21 fins compared to 65 fins for the micro-fin tube). However, the difference in the performance of the two tubes was especially evident in the pressure drop penalty factors, which were twice as large for the low-fin tube--1.3 for the micro-fin tube compared to 2.6 for the low-fin tube.

The performance of different micro-fin tube geometries during evaporation and condensation of HCFC-22 was evaluated in several recent studies by Schlager et al. For example, three different 9.5-mm OD micro-fin tubes were compared by Schlager et al. (1989a), while three different 12.7-mm OD tubes were compared by Schlager et al. (1989b). This latter study also compared the heat transfer performance of two tubes of different diameter, 9.5 mm and 12.7 mm, at similar mass fluxes. For each tube, an average heat transfer coefficient was obtained for a 90%

quality change occurring in a 3.7-m tube. In the first study, the main difference between the three 9.5-mm tubes was that the spiral angles were varied, being 15°, 18°, and 25°. The heat transfer enhancement factors varied from 1.4 to 1.9, while pressure drop penalty factors varied from 1.0 to 1.4. The performance of the three tubes was similar, falling within the band of experimental uncertainties. However, the tube with a 25° spiral angle appeared to be slightly better for evaporation and the 18° spiral angle tube was slightly better for condensation.

The second study (Schlager et al. 1989b), for the 12.7-mm-diameter tube, produced results that were similar to those for the smaller 9.5-mm-diameter tube. In fact, a general conclusion made by Schlager is that enhancement factors measured for one particular diameter might then be applied to other diameters. Another observation made was that the heat transfer enhancement factors exceeded the area ratios (the ratio of the micro-fin tube's surface area to the area of a smooth tube of equivalent diameter), which varied from 1.38 to 1.55 for nearly all test conditions. Interestingly, as the mass flux was increased, the heat transfer enhancement factor values approached the area ratio. A possible explanation is that at high mass fluxes, the enhancement in heat transfer with the micro-fin tube is due to the increase in the area, with the turbulence being so high that additional disturbances caused by the fins do not significantly add to the heat transfer. In contrast, at low flow rates, the presence of the fins causes large disturbances in the flow, which, in turn, results in significant heat transfer enhancement over that caused by the area increase.

A series of recent publications have resulted from an ASHRAE-sponsored research project, RP-469, which dealt with heat transfer and pressure drop of HCFC-22 and oil mixtures in both smooth and micro-fin tubes. The results of this work were reported by Schlager et al. (1989c through 1989e); a summary of each reference is shown in Table 1. The heat transfer data reported in each case was for a 3.67 m (12 ft) long by 9.52 mm (0.375 in) OD tube, with the quality varying from 15% to 85% from the inlet to exit, respectively (or from 85% to 15% for condensation studies). The micro-fin tube tested had an 18° spiral angle, 60 fins, a fin height of 0.2 mm (.008 in), and an area ratio of 1.5. In chronological order, a brief overview of each study is presented.

In Schlager et al. (1989c), the performance of the micro-fin tube described above was evaluated for a mixture of HCFC-22 and a naphthenic oil with a viscosity of 150 SUS. Small quantities of the lubricant, about 1.5%, increased the evaporation heat transfer coefficient by about 11%. This enhancement decreased as the mass flux increased and as additional oil was added. For example, at one mass flux, the enhancement factor varied from 2.4 for pure refrigerant to 1.9 for 5% oil concentration. For condensation, oil addition decreased the heat transfer coefficient in the micro-fin tube by as much as 16%. However, the heat transfer enhancement factor for condensation was not affected by the oil concentration. Depending on the mass flux, the heat transfer enhancement factor varied from 1.9 to 2.4.

Schlager et al. (1989d) reported the results of a study of oil viscosity effects on heat transfer in a micro-fin tube. Unlike the 150 SUS oil reported in the previous paragraph, a higher viscosity 300 SUS naphthenic oil did not enhance heat transfer during evaporation. Rather, heat transfer decreased for all oil concentrations, with the maximum decrease being 30% at a 5% oil concentration. For condensation, the micro-fin tube performance was similar for both oil viscosities.

The micro-fin tube reported above was compared to a low-fin tube (21 fins per tube, 30° spiral angle, 0.38-mm fin height, and a 1.8 area ratio) in Schlager et al. (1989e). For evaporation of pure refrigerant, the heat transfer enhancement factor was similar for both tubes, varying from 2.3 to 1.8. However, the pressure drop penalty factors were about 10% to 30% smaller for the micro-fin tube (this was also the case for condensation). The higher pressure drops in the low-fin tube could be caused by either the larger spiral angle, the larger area ratio, or a combination of both. For condensation, the enhancement factors for the micro-fin tube were much higher than that for the low-fin tube. For example, at a particular flow rate, they were 2.3 compared to 1.9. Of special note, addition of oil degraded the performance of the low-fin tube significantly more than that of the micro-fin tube for evaporation.

A series of two papers by Schlager et al. (1990a, b) focused on an approach for predicting the performance of micro-fin tubes when operating with refrigerant-oil mixtures. Because a detailed study evaluating the effects of geometrical parameters has not been performed, it was not possible to derive a general equation that can be used to design micro-fin tubes. Rather, geometry-specific equations for heat transfer and pressure drop were developed for both pure refrigerants and oil-refrigerant mixtures.

Insight into how micro-fin tubes improve heat transfer was provided by Manwell and Bergles (1990). Specifically, they investigated flow patterns in micro-fin tubes by using still photography to record typical flow patterns. By comparing flow-pattern maps for a smooth tube with photographs of a micro-fin tube, they concluded that the presence of the spiral flow reduces the stratified flow region. In other words, at low flow rates, where stratified flow might otherwise occur, the spiraled fins cause the upper surface to be wetted, thus producing a thin liquid film on the grooved wall. Since this upper wall is dry in a smooth tube during stratified flow, the presence of the liquid film enhances heat transfer. This observation is consistent with the results of experimental studies by Schlager et al. (1989c through 1989e), which showed that enhancement factors for micro-fin tubes are much higher at low flow rates.

TEST FACILITIES

The test facility used in this study is described in detail in a companion paper by Eckels and Pate (1991). This facility was designed for the purpose of measuring in-tube heat transfer coefficients during both evaporation and condensation of refrigerants. A schematic diagram of the test facility is shown in Figure 1. In addition to a test section, other components set the condition of the refrigerant and water entering the test section. A brief description of the major components of the loop is given below.

Test Section

The test section contains the tube undergoing experimental testing. For this study, the tube was 3.67 m (12 ft) long with an OD of 9.52 mm (0.36 in). Two tubes were tested during this study--a smooth tube and a micro-fin tube. The smooth tube had a 9.52 mm (0.36 in) OD and an ID of 8.0 mm (0.314 in). The micro-fin tube also had a 9.52-mm (0.36 in) OD and a maximum inside diameter of 8.72 mm (0.314 in). In addition, the micro-fin tube had an 17° spiral angle, 60 fins, a fin height of 0.2 mm (.008 in), and an area ratio of 1.5. A diagram of the micro-fin tube that was tested is shown in Figure 2. Flowing around the outside of the test tube in an annulus is water that is conditioned by a separate water flow loop. Temperatures and pressures are measured on the refrigerant side at the inlet and exit of the test tube, while the temperature of the water entering and exiting the test section is also measured. The experimental uncertainties are listed in Table 2.

Refrigerant Loop

The refrigerant loop sets the condition of the refrigerant entering the test section. The loop contains a positive-displacement pump, a bladder accumulator, heaters, and an after-condenser. The prime mover is a liquid pump rather than a gas (or vapor) compressor, which has the advantage of not requiring the addition of a lubricating oil to the refrigerant system.

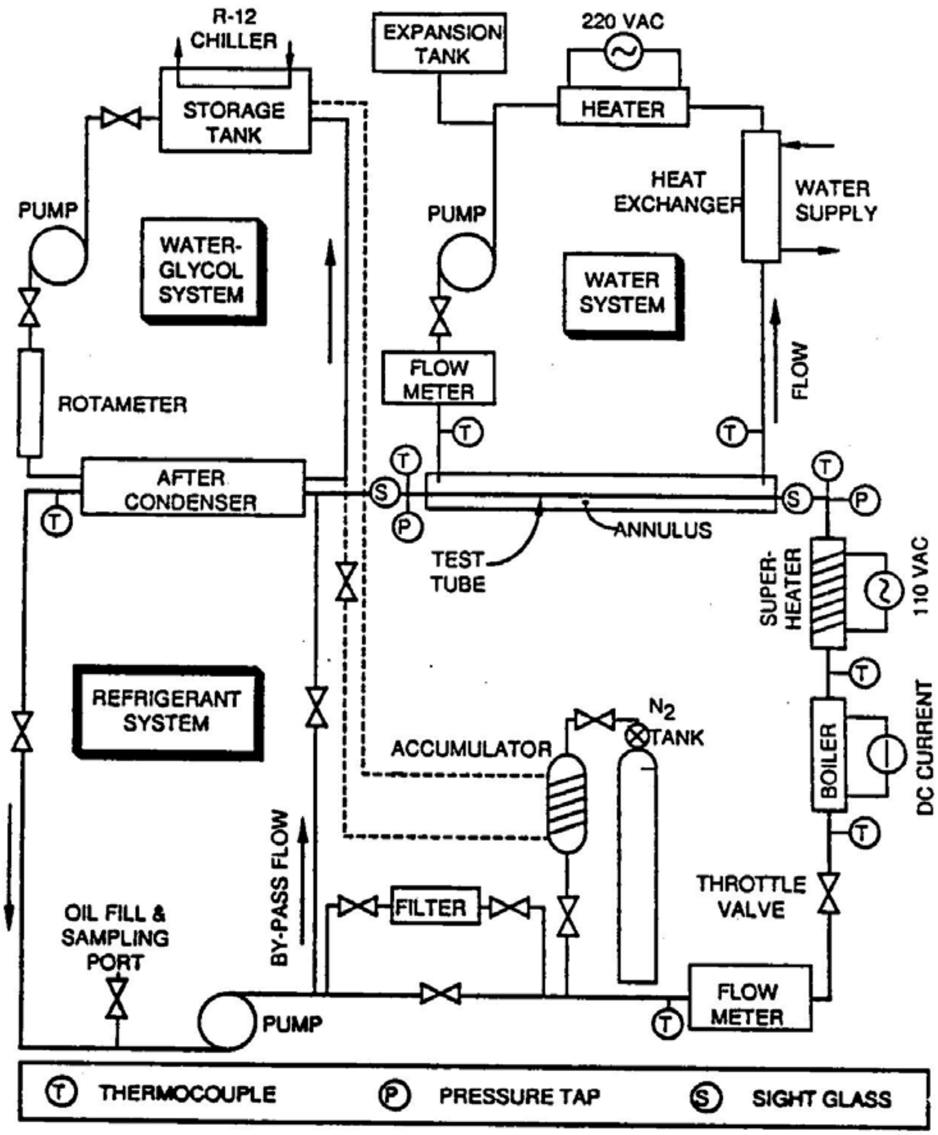

Figure 1 Schematic drawing of test facility

Table 2 Experimental Uncertainties For Sensor (From Manufacturers' Literature)

Temperatures (at inlet and exit of test section)	± 0.2°C (± 0.36°F)
Absolute Pressures (refrigerant)	± 9 kPa (± 1.3 psi)
Pressure Differentials (refrigerant)	± 0.2 kPa (± 0.03 psi)
Refrigerant Flow Rate	± 1%
Water Flow Rate	± 1%

Pressure is controlled in the refrigerant loop and, hence, the test section by the bladder accumulator. The heaters (a boiler and a superheater) set the quality entering the test section. The refrigerant flow rate is measured by a positive-displacement flowmeter (see Table 2 for experimental uncertainties).

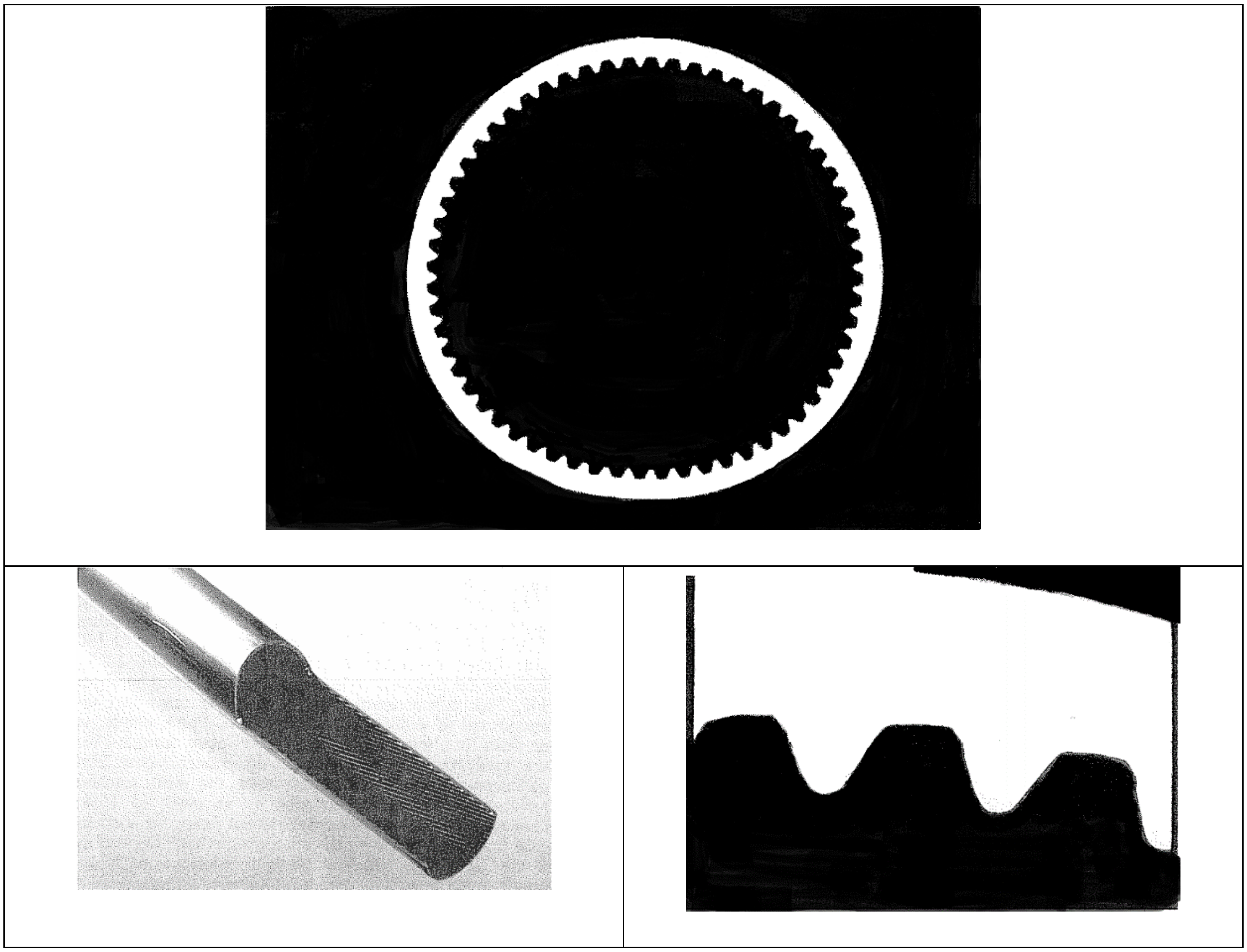

Figure 2 Photograph of the micro-fin tube

Water Loop

The water loop sets the condition of the water entering the annulus side of the test section. A centrifugal pump circulates the water, a magnetic flowmeter measures the flow rate, and a combination of a heat exchanger and an electrical heater controls the water temperature. Water-Glycol Loop

The after-condenser, which condenses and subcools the refrigerant exiting the test section, is cooled by a water-glycol flow loop. This loop consists of a 17.5-kW (5-ton) refrigeration unit whose evaporator is interfaced to a 209-L (55-gal) storage tank containing a 50/50 mixture of water and glycol.

DATA ACQUISITION AND REDUCTION PROCEDURES

Data acquisition is performed with a personal computer, a 40-channel scanner, and a multimeter. The temperature, pressure, and flow rate data from the sensors, described previously, are measured by the data acquisition system and then used to calculate heat transfer coefficients, qualities, and mass fluxes. A detailed description of the equations used to calculate the above parameters is presented in a companion paper by Eckels and Pate (1991).

The equation used to obtain the in-tube heat transfer coefficient is

$$h_i = \frac{1}{\left(\frac{1}{U_o} - \frac{1}{h_o}\right)\frac{A_i}{A_o}}. \qquad (1)$$

The overall heat transfer coefficient, U_O, is obtained from energy balance measurements on the water side of the test section by using a log mean temperature difference approach. The inside surface area for the micro-fin tube used in Equation 1 is calculated from an equivalent-diameter smooth tube assumption based on the maximum inside diameter of the micro-fin tube. The outside heat transfer coefficient used in Equation 1 is obtained from a Wilson plot technique. Entering and exiting qualities are obtained from energy balances on the test section and upstream heaters. The mass flux of the refrigerant in the test tube is calculated from the mass flow rate and an average cross-sectional area. For the smooth tube, the inside radius of the tube is used to calculate the cross-sectional area. For the micro-fin tube, the average cross-sectional area is calculated from the maximum inside radius minus half the fin height. The experimental uncertainties in the measured parameters are listed in Table 3.

Table 3: Experimental Uncertainties For Parameters Of Interest (Based On Uncertainty Analysis Or Statistical Analysis)

Heat Transfer Coefficients (calculated)	
mass flux of 130 kg/m²s (95,860 lb/ft²·h)	
evaporation	± 14%
condensation	± 13%
mass flux of 400 kg/m²s (294,000 lb/ft²·h)	
evaporation	± 9%
condensation	± 8%
Mass Flux, Quality, and Pressure Drop (calculated)	
refrigerant mass flux	± 3%
refrigerant quality	± 3.5%
pressure drop	± 2 kPa (0.29 psi)

TEST CONDITIONS

The heat transfer coefficients and pressure drops of HFC-134a and CFC-12 were measured in a micro-fin and smooth tube at three evaporation and three condensation temperatures. At each temperature, a range of mass fluxes were tested for the micro-fin tube and smooth tube. The range of experimental conditions is listed in Table 4.

Eckels, S.J. and M.B. Pate, *Evaporation and condensation of HFC-134a in a smooth tube and a micro-fin tube.* ASHRAE Transactions, 1991. **97**(2): p. 68-78.

Table 4 Test Conditions

	Evaporation	**Condensation**
Average Temperature °C	5 - 15	30-50
Mass flux (kg/m² s)	125 - 400	125 - 400
Quality in (%)	5-13	80 -88
Quality out (%)	80 - 88	5 - 13

RESULTS FOR HFC-134a

Evaporation

Heat Transfer

Figure 3 shows evaporation heat transfer coefficient data for HFC-134a for both the smooth tube and the micro-fin tube. In both cases, the heat transfer coefficients increase with mass flux and with temperature. For example, as the mass flux is varied over a 1:3 range, then heat transfer coefficients increase by about 50% for the micro-fin tube and by about 100% for the smooth tube. The effect of temperature on heat transfer coefficients for both tubes is similar and less than 30% over a wide range of mass fluxes.

Another observation from Figure 3 is that the heat transfer coefficients are significantly higher for the micro-fin tube compared to the smooth tube. For example, at the lowest mass flux they are as much as 150% higher, while at the highest mass flux they are about 50% higher.

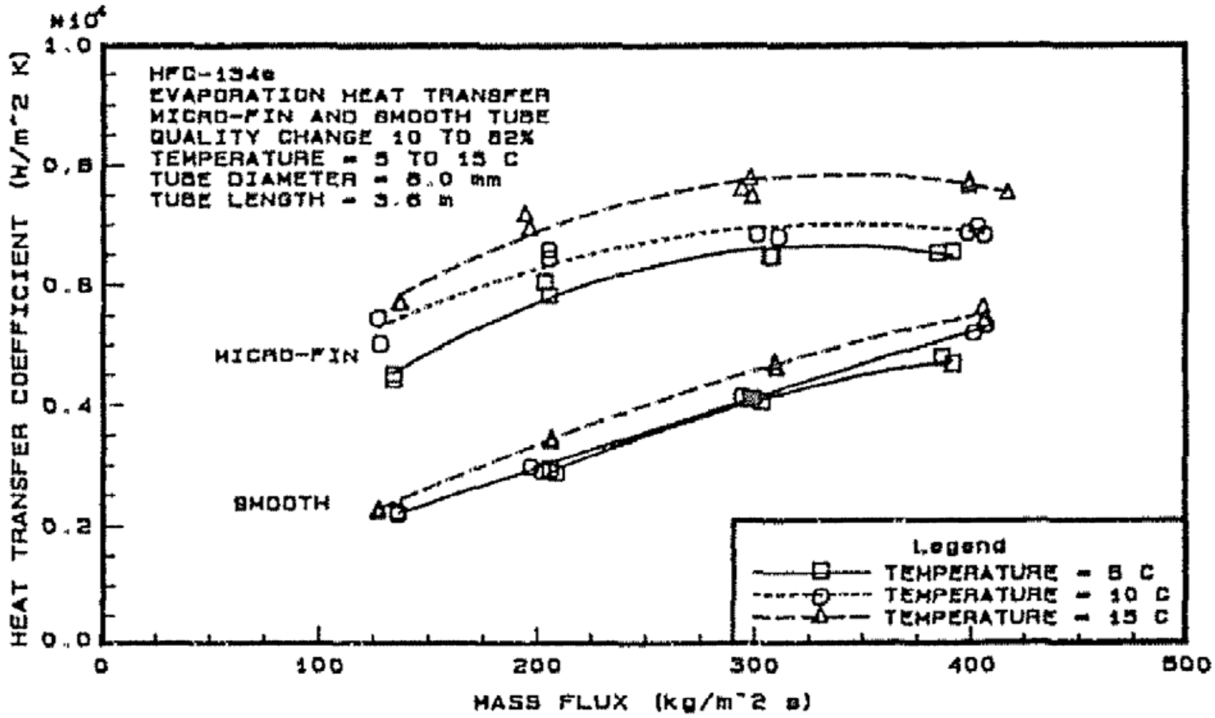

Figure 3: HFC-134a evaporaiton heat transfer coefficients in a smooth tube adn micro-fin tube

Pressure Drop

Pressure drop data for HFC-134a in smooth and micro-fin tubes are shown in Figure 4. For both tubes, the pressure drop of the refrigerant as it flows through the tube increases significantly with mass flux. For example, as the mass flux is tripled, the pressure drop increases by as much as a factor of 10. Also shown in Figure 4 is that pressure drops are also a function of refrigerant temperature. This is especially true for the micro-fin tube, where pressure drops for specific mass fluxes can vary by as much as 40% for a temperature change of 5°C to 15°C.

Enhancement and Penalty Factors

Another approach for comparing the micro-fin tube heat transfer performance with that of the smooth tube is to form heat transfer enhancement factors, EF, defined as the ratio of the micro-fin tube heat transfer coefficient to that of a comparable smooth tube at a similar mass flux, heat flux, pressure level, and inlet and outlet quality. Heat transfer enhancement factors for HFC-134a are shown in Figure 5. Pressure drop performance comparisons between the micro-fin tube and smooth tube can be made by forming ratios of pressure drops in a manner similar to that used to form heat transfer enhancement factors. These ratios are hereafter referred to as pressure drop penalty factors, PF.

`Figure 5 shows both heat transfer enhancement factors, EF, and pressure drop penalty factors, PF, for the micro-fin tube with HFC-134a. The EFs vary from 2.5 a low mass flux to about 1.5 at a high mass flux. Interestingly, the value of 1.5 at the high mass flux corresponds to the area ratio for the micro-fin tube. As discussed previously, a possible explanation is that at the high mass flux, the heat transfer increase in the micro-fin tube is due to the area increase. Also observable in Figure 5 is the fact that enhancement factors do not vary with temperature, signifying that temperature has about the same effect on smooth and micro-fin tubes.

.

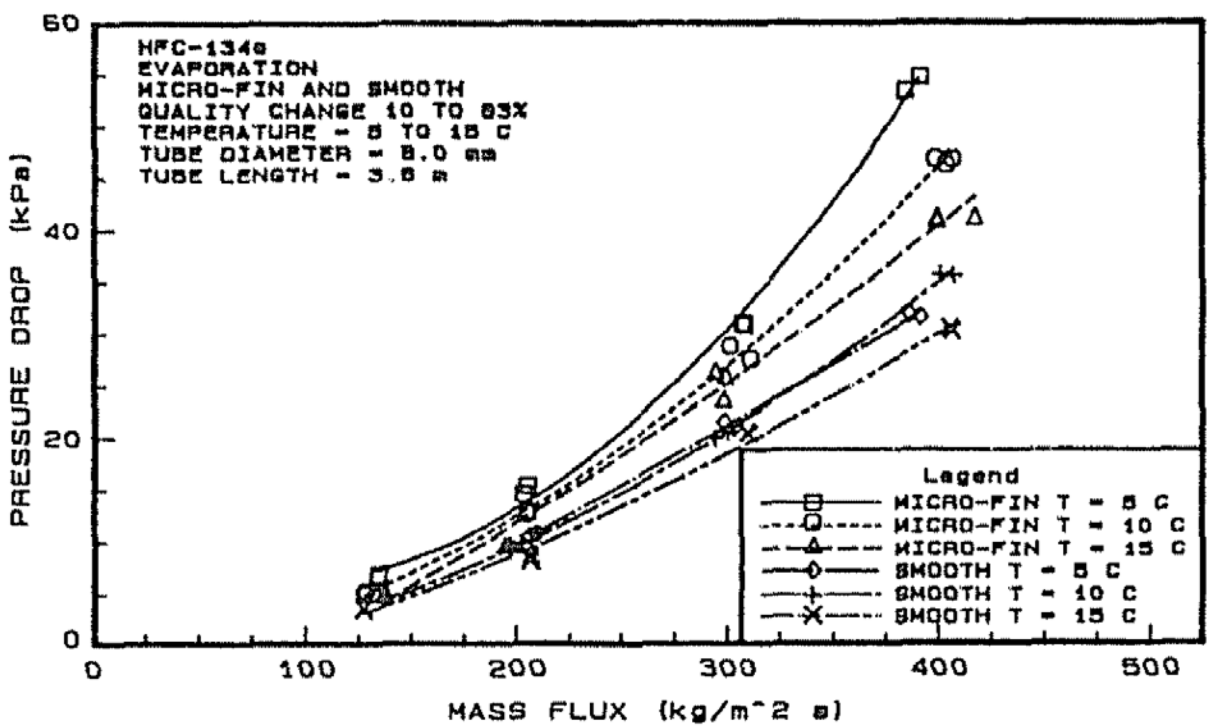

Figure 4: HFC-134a evaporation pressure drop in a smooth tube and micro-fin tube

Eckels, S.J. and M.B. Pate, *Evaporation and condensation of HFC-134a in a smooth tube and a micro-fin tube.* ASHRAE Transactions, 1991. **97**(2): p. 68-78.

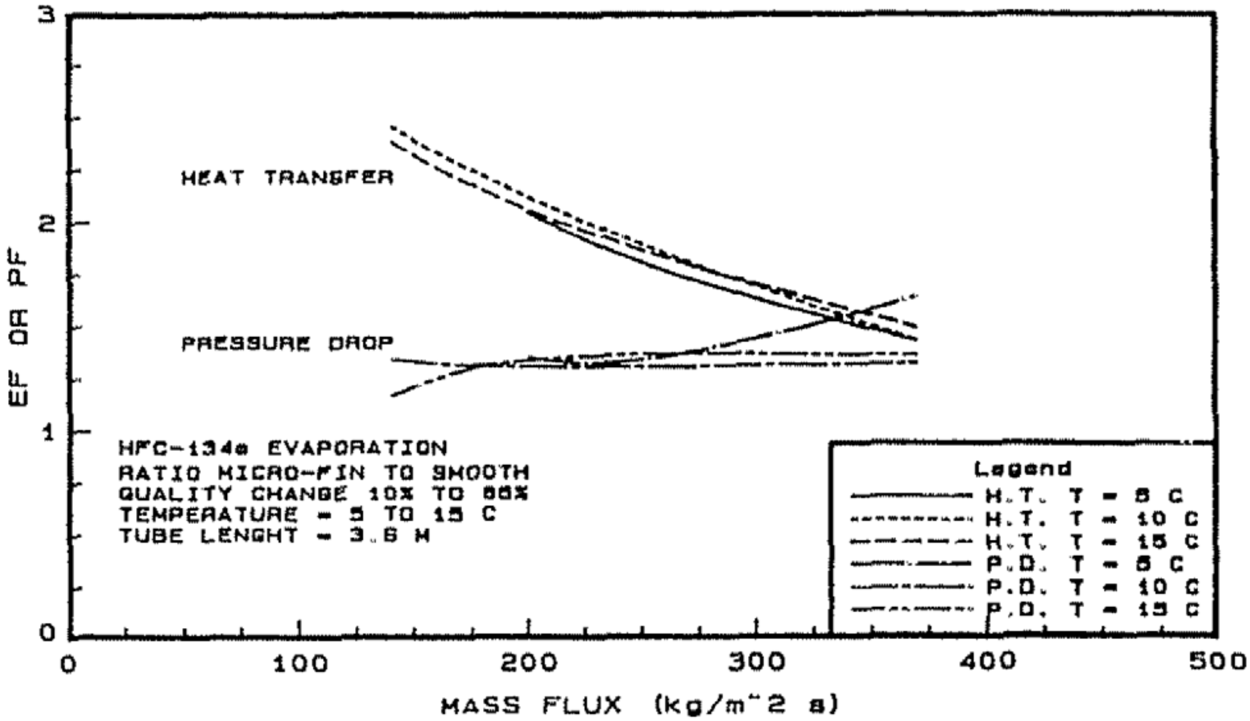

Figure 5: Heat Transfer enhancement factor and pressure drop penalty factor for evaporaiton of HFC-134a

The PFs shown in Figure 5 are relatively constant with mass flux with a value of 1.3 being representative; however, they vary considerably--from about 1.0 to 1.6--with refrigerant temperature. Even though for most conditions the PF does not vary with temperature there appears to be an increase at the highest mass flux for the lowest temperature data, namely 5 oC. In fact, the penalty factors increase so much as the evaporation temperature is reduced that at the lowest temperature and highest mass flux the enhancement factor is less than the penalty factor. This crossover has not been observed in past studies of micro-fin tubes and is difficult to explain with the present understanding of fundamental evaporation phenomena in micro-fin tubes. Additional studies are necessary to explain this behavior.

Condensation

Heat Transfer

Condensation heat transfer data for the smooth tube and micro-fin tube with HFC-134a are shown in Figure 6. In both cases, the heat transfer coefficients increase with mass flux and temperature. In the case of the micro-fin tube, the effect of temperature is appreciable, with heat transfer coefficients increasing by as much as 30% as the temperature is reduced from 50°C to 30°C. The effect of mass flux on heat transfer coefficients can be observed by the fact that the coefficients are increased by about 30% for the micro-fin tube and by about 50% for the smooth tube as the mass flux is increased by a factor of three.

Another observation from Figure 6 is that heat transfer coefficients are significantly higher for the micro-fin tube compared to the smooth tube. At the lowest and highest mass fluxes, they are about 110% and 70% higher, respectively.

Eckels, S.J. and M.B. Pate, *Evaporation and condensation of HFC-134a in a smooth tube and a micro-fin tube.* ASHRAE Transactions, 1991. **97**(2): p. 68-78.

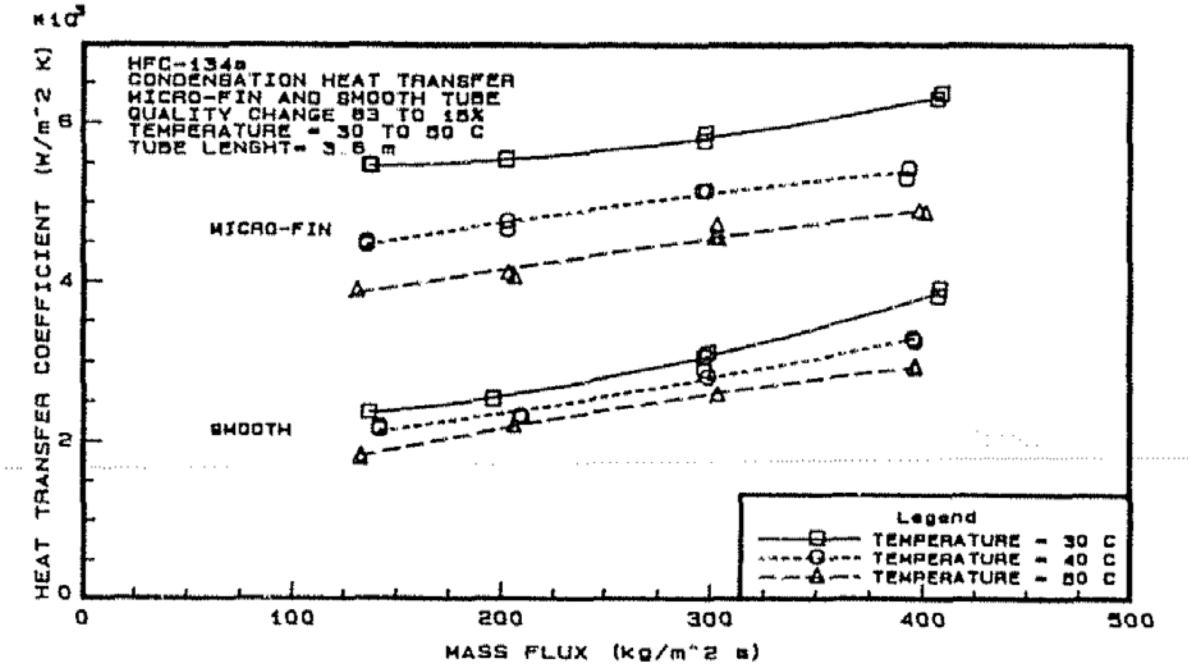

Figure 6 Condensation heat transfer coefficients in a smooth tube and micro-fin tube

Pressure Drop

Pressure drop data during condensation of HFC-134a for smooth tubes and micro-fin tubes are shown in Figure 7. As with heat transfer coefficients, the pressure drop varies considerably with mass flux and temperature for both tubes. Also shown is the fact that the pressure drop in the micro-fin tube is larger than that in the smooth tube. For example, at 40°C, the pressure drop is about 90% higher at a low mass flux and about 30% higher at a higher mass flux. Caution must be used when interpreting results for condensation pressure drops at low mass fluxes because the relative uncertainties in the pressure drop are on the order of the results.

Enhancement and Penalty Factors

Heat transfer enhancement factors and pressure drop penalty factors for condensation as a function of mass flux are shown in Figure 8. Enhancement factors vary from a maximum of about 2.5 at low mass fluxes to a minimum of 1.75 at high mass fluxes. Penalty factors follow the same general trend as enhancement factors in that they decrease as both the temperature and mass flux increase. Over nearly all of the flow condition range, the penalty factors are less than the enhancement factors. For example, at 40°C at a mid-flow range, the PF is 1.5, while the EF is higher at 2.0. The one exception to this trend is that, at the lowest mass flux at 30°C, the penalty factor is greater than the enhancement factor. However, the large uncertainties in pressure drops at low mass fluxes preclude any general conclusions.

RESULTS FOR CFC-12

Heat transfer and pressure drop measurements of CFC-12 are important because existing refrigeration systems have been designed for use with CFC-12. Therefore, if modifications to these designs are to be made to accommodate HFC-134a use, then the relative performance of HFC-134a and CFC-12 becomes important. In this section, CFC-12 data are reported for evaporation and condensation heat transfer and, in the next section, a comparison is made between the two refrigerants.

Eckels, S.J. and M.B. Pate, *Evaporation and condensation of HFC-134a in a smooth tube and a micro-fin tube.* ASHRAE Transactions, 1991. **97**(2): p. 68-78.

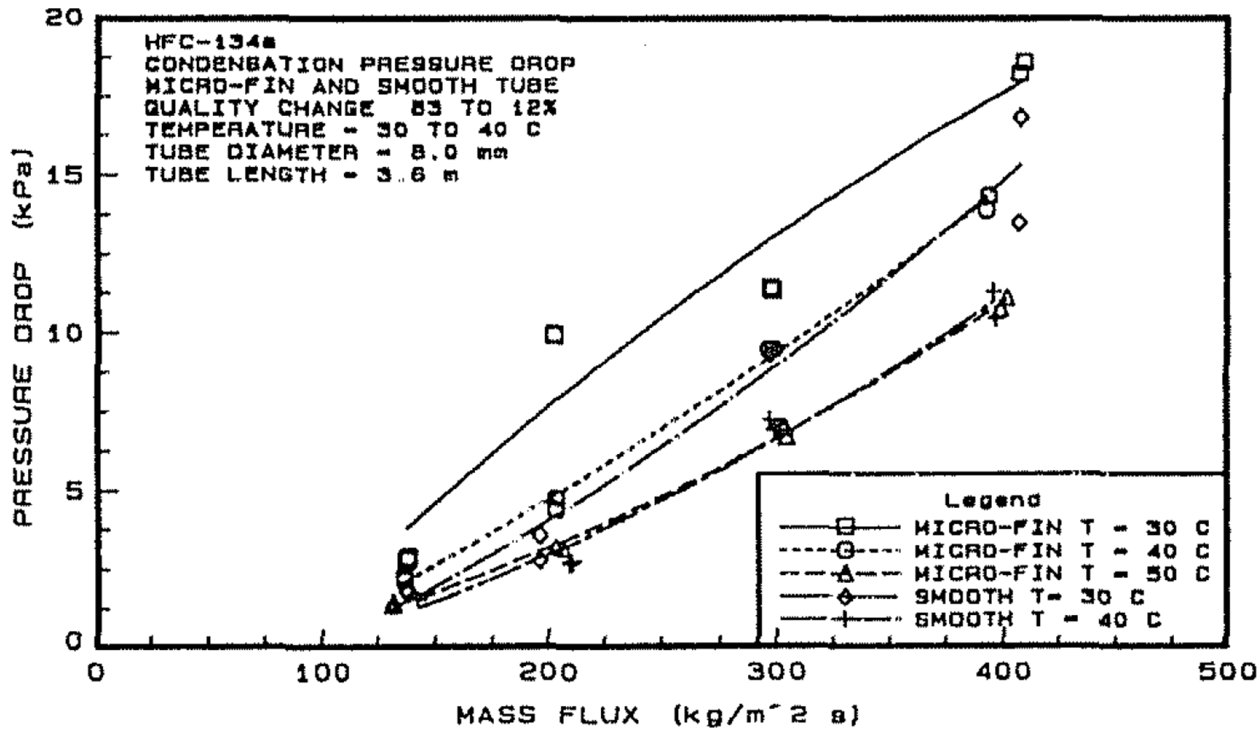

Figure 7 HFC-134a condensaiton pressure drops in a smooth tube and micro-fin tube.

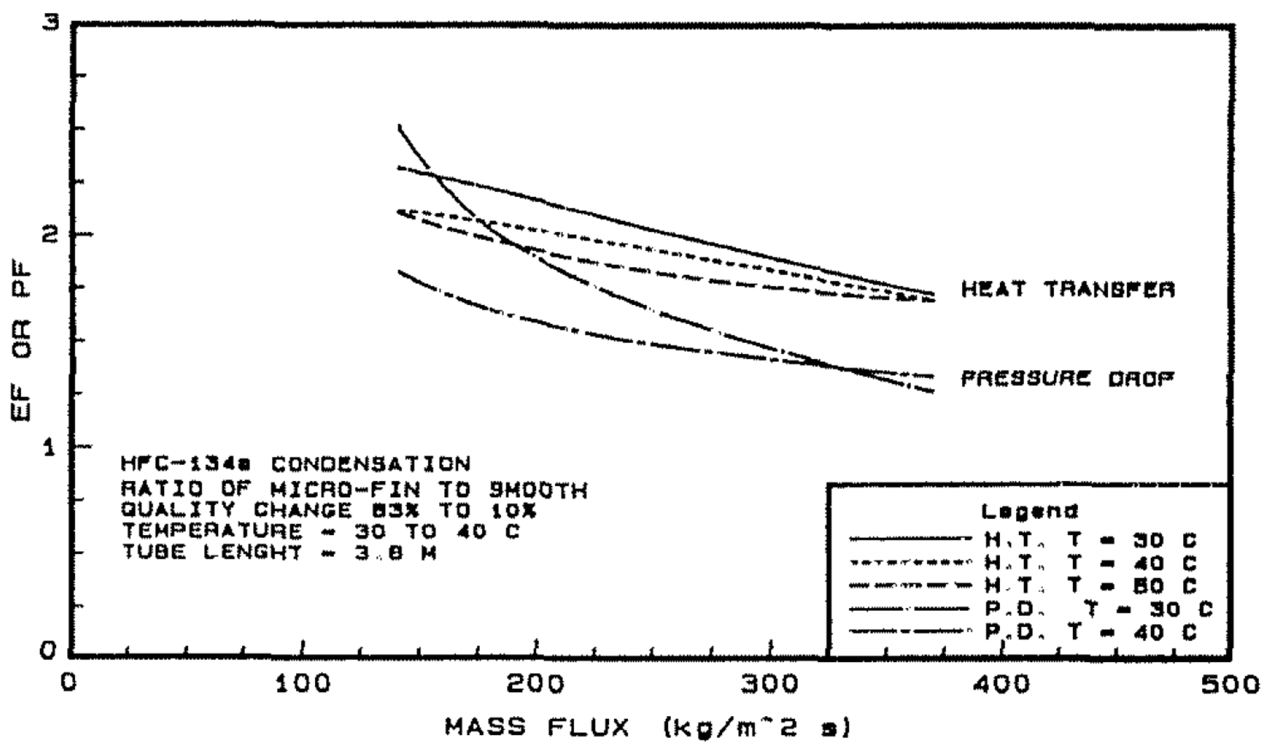

Figure 8 Heat transfer enhancement factor and pressure drop penalty factor for condensation of HFC-134a

Evaporation

Heat Transfer

Figure 9 shows evaporation heat transfer coefficient data for CFC-12 for both smooth and micro-fin tubes. The trends are similar to those observed for HFC-134a in that the coefficients increase with temperature and mass flux. Again, the heat transfer coefficients for the micro-fin tube are more than 100% larger than those for the smooth tube.

Pressure Drop

Pressure drop data for CFC-12 during evaporation are shown in Figure 10. For both tubes, pressure drops increase with mass flux and decrease with increasing evaporation temperature. The pressure drops for the micro-fin tube are 10% to 50% larger compared to the smooth tube.

Enhancement and Penalty Factors

Heat transfer enhancement factors, EF, and pressure drop penalty factors, PF, for CFC-12 are shown in Figure 11. The enhancement factor varies with mass flux, however, it is not a function of evaporation temperature. For example, it varies from 3.0 at low mass fluxes to about 1.7 at high mass fluxes. There are no noticeable trends in the penalty factor with changes in mass flux and temperature. However, values of PF vary from a maximum of 2.0 to a minimum of 1.5. In all cases, the increase in heat transfer due to micro-fins is greater than the increase in pressure drop.

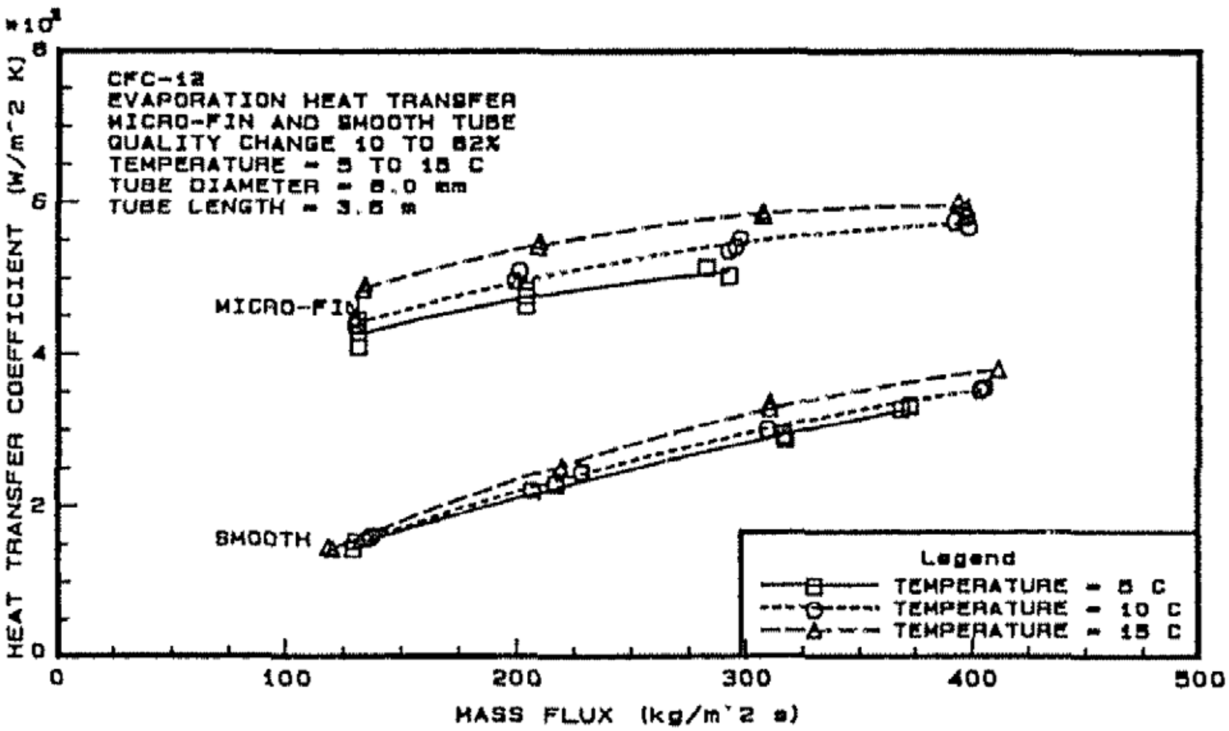

Figure 9 CFC-12 evaporation heat transfer coefficient in a smooth tube and micro-fin tube

Eckels, S.J. and M.B. Pate, *Evaporation and condensation of HFC-134a in a smooth tube and a micro-fin tube.* ASHRAE Transactions, 1991. **97**(2): p. 68-78.

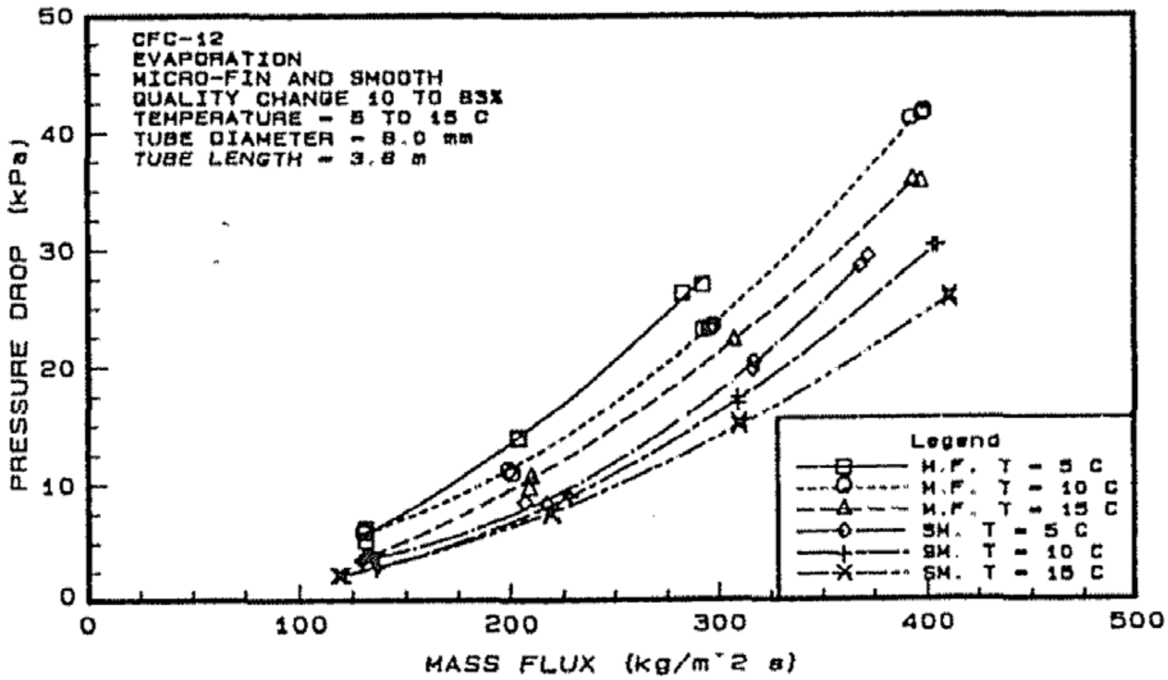

Figure 10 CFC-12 evaporaiton pressure drops in a smooth tube and micro-fin tube

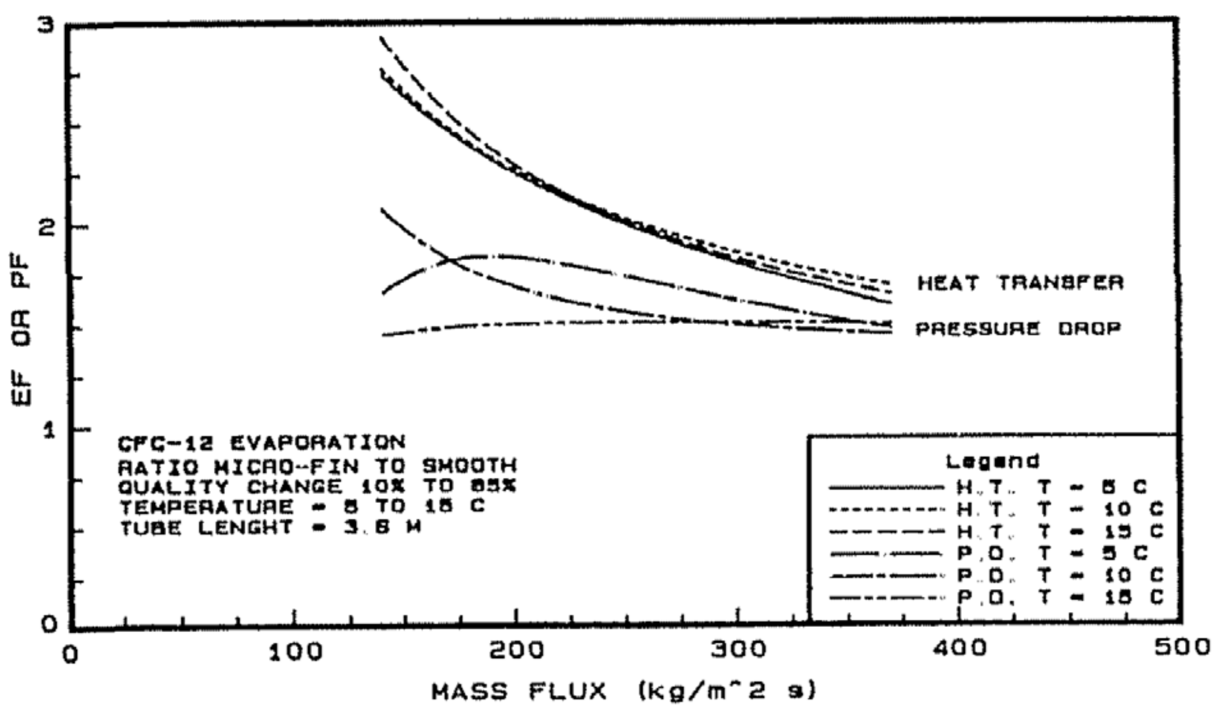

Figure 11 Heat transfer enhancement factor and pressure dorp penalty factor for evaporation of CFC-12

Condensation

Heat Transfer

Condensation heat transfer coefficients for CFC-12 in a micro-fin tube and a smooth tube are shown in Figure 12. The trends shown are comparable to those observed previously in Figure 6 for HFC-134a. As with HFC-134a, the CFC-12 heat transfer coefficients are significantly higher for the micro-fin tube than for the smooth tube. For example, at 40°C and a mid-range mass flux, the heat transfer coefficients for the micro-fin tube are about 100% larger than the smooth tube.

Pressure Drop

Pressure drops for CFC-12 during condensation are shown in Figure 13. As with heat transfer, the overall trends are similar to those observed for HFC-134a.

Enhancement and Penalty Factors

Heat transfer enhancement factors, EF, for CFC-12 are shown in Figure 14. The EFs vary from a maximum of 2.3 at a low mass flux and 30°C condensing temperature to a minimum of 1.7 for all three condensing temperatures at the highest mass flux. Pressure drop penalty factors, PF, are also shown in Figure 14. For any given condensation temperature, penalty factors are less than the enhancement factors. There appears to be an overall trend of increasing enhancement factors and penalty factors with decreasing condensing temperature.

COMPARISON OF HFC-134a AND CFC-12

Smooth Tube

Experimental heat transfer coefficients for HFC-134a and CFC-12 can be compared by forming the ratio of heat transfer coefficients (i.e., the heat transfer coefficient for HFC-134a is divided by the coefficient for CFC-12) at the same mass flux. Another method of comparing the

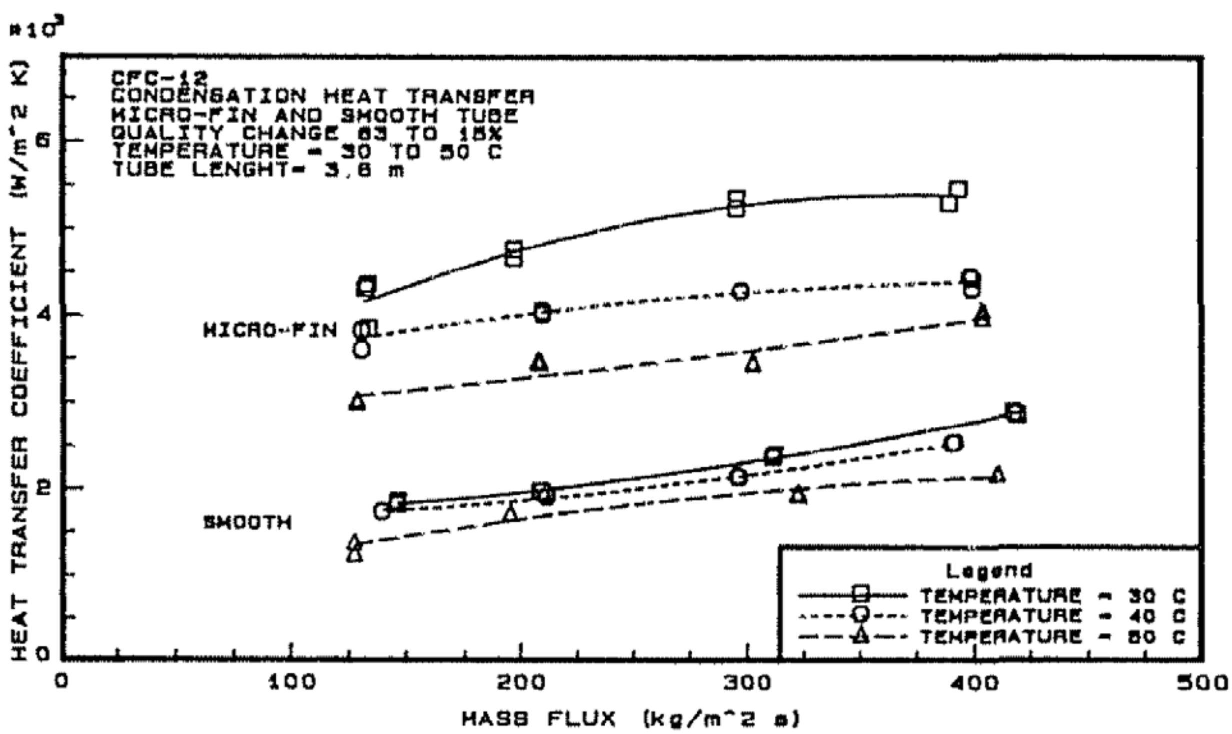

Figure 12: CFC-12 condensation heat transfer coefficients in a smooth tube and micro-fin tube

Eckels, S.J. and M.B. Pate, *Evaporation and condensation of HFC-134a in a smooth tube and a micro-fin tube.* ASHRAE Transactions, 1991. **97**(2): p. 68-78.

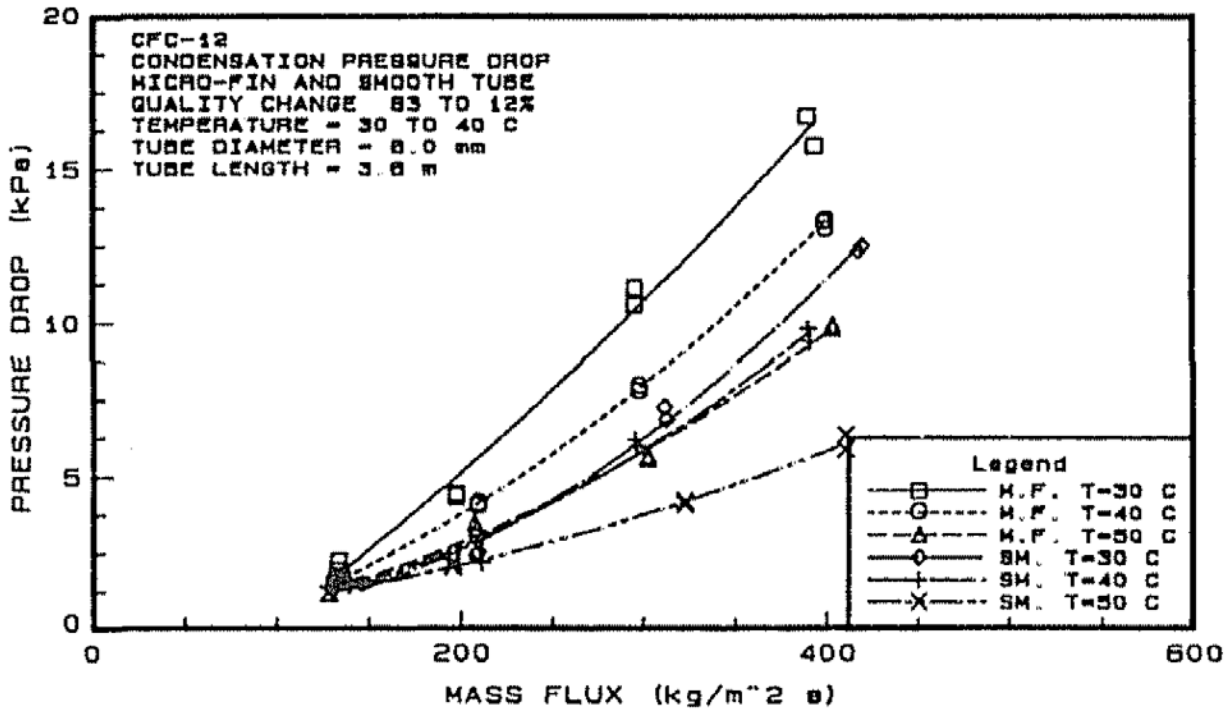

Figure 13 Condensation pressure drops in a smooth tube and micro-fin tube

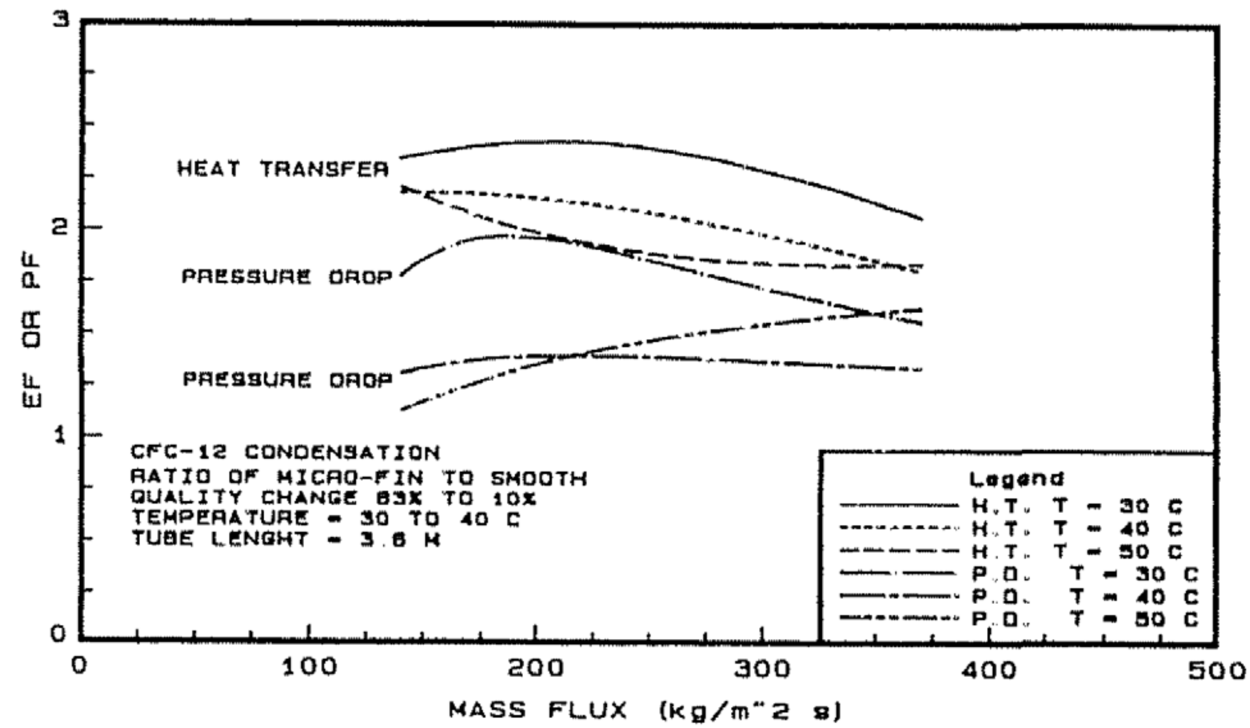

Figure 14 Heat Transfer enhancement factor and pressure drop penalty factor for condensation of CFC-12

Table 5 Comparison of HFC-134a and CFC-12 for Smooth Tube

	Same Mass Flux	**Same Cooling (Heating) Capacity**
Heat Transfer		
· evaporation	1.4*	1.1
· condensation	1.25	1.1
Pressure Drop		
· evaporation	1.0 - 1.5	0.7 - 0.9
· condensation	1.0 - 1.5	0.6 - 0.9

*HFC-134a to CFC-12 ratios

coefficients for the two refrigerants is to compare them at similar cooling capacities (or heating capacities in the case of condensation). This second comparison is based on the fact that for the same cooling capacity, a lower flow rate of HFC-134a is required compared to the flow rate of CFC-12. This reduced flow rate is the result of the enthalpy of vaporization being higher for HFC-134a compared to CFC-12.

Ratios of heat transfer coefficients and pressure drops for the two refrigerants at both similar mass fluxes and similar cooling capacities (or heating capacities) are listed in Table 5. The ratios for heat transfer are relatively constant; however, the ratios vary considerably for pressure drop, depending on the mass flux and temperature. For evaporation at similar mass fluxes, the HFC-134a to CFC-12 heat transfer ratio is 1.4, which signifies that the coefficient for HFC-134a is about 40% higher than that for CFC-12. When the flow rate of HFC-134a is reduced (or else the flow rate of CFC-12 is increased) to achieve an equivalent cooling capacity comparison, the heat transfer coefficient of HFC-134a is reduced. However, even then the heat transfer coefficient is still higher than the CFC-12 value. For example, the ratio shown in Table 5 is 1.1 for an equivalent cooling capacity, which signifies that the HFC-134a coefficient is 10% higher than the CFC-12 value. Similar results can be observed for condensation in that the ratio for similar mass fluxes is 1.25, while the ratio for similar heating capacities is 1.1.

The HFC-134a to CFC-12 ratios for pressure drops are also shown in Table 5. As mentioned previously, the ratios in Table 5 for pressure drops vary and, as such, a range of values has been listed. At similar mass fluxes, the pressure drop for HFC-134a for both evaporation and condensation is higher than that for CFC-12, as evidenced by ratios that range from 1.0 to 1.5.

However, when the mass flux of HFC-134a is decreased to achieve an equivalent heating or cooling capacity (or else the mass flux of CFC-12 is increased), then the pressure drop of HFC-134a relative to CFC-12 is decreased. In fact, the range of ratios from 0.6 to 0.9 shown in Table 6 signifies that the pressure drop of HFC-134a is less than the pressure drop of CFC-12. The fact that the heat transfer coefficients are larger while the pressure drops are lower for HFC-134a could potentially be used to design HFC-134a heat exchangers that have a higher thermal performance compared to CFC-12 heat exchangers.

Micro-Fin Tube

The relative performance of micro-fin tubes operating with HFC-134a and CFC-12 can be determined by comparing heat transfer enhancement factors, EF, and pressure drop penalty factors, PF, for the two refrigerants. As with the comparison for the smooth tubes presented in the previous section, the comparison for the micro-fin tube is performed at similar mass fluxes and at an

Table 6 Comparison of Heat Transfer Enhancement Factors for HFC-134a and CFC-12

		Similar Mass Flux Comparison		**Similar Cooling (Heating) Capacity Comparison**	
		EF for HFC-134a	EF for CFC-12	EF for HFC-134a	EF for CFC-12*
Evaporation T = 10°C	low mass flux 150 kg/m²s	2.3	2.6	2.3	2.3
	high mass flux 300 kg/m²s	1.7	1.9	1.7	1.8
Condensation T = 40°C	low mass flux 150 kg/m²s	2.2	2.2	2.2	2.2
	high mass flux 300 kg/m²s	1.9	2.0	1.9	1.8

*Enhancement factors for CFC-12 are taken at mass fluxes that are increased by 30% (equivalent to the ratio of the enthalpy of vaporizations for HFC-134a and CFC-12).

equivalent cooling (or heating) capacity. Heat transfer enhancement factors are presented in Table 6, while pressure drop penalty factors are presented in Table 7.

As shown in Table 6, the EF for evaporation of HFC-134a is slightly lower than that of CFC-12, being 2.3 compared to 2.6 at low mass fluxes and 1.7 compared to 1.9 at higher mass fluxes. However, when the mass flux of CFC-12 is increased to achieve an equivalent cooling capacity, then the enhancement factor of CFC-12 is reduced. This behavior of EF with mass flux is consistent with observations made in other sections of this paper. As a result of reduced EFs, the enhancement factors of HFC-134a and CFC-12 are comparable at an equivalent cooling capacity.

Also shown in Table 6 are EFs for condensation heat transfer. A general observation is that the enhancement factors are similar for HFC-134a and CFC-12 for both the similar mass flux case and for the equivalent heating capacity case. For example, for both cases at a low mass flux, the EF for both refrigerants is 2.2, while at high mass fluxes the EF is about 1.8 to 2.0.

A comparison of pressure drop penalty factors, PF, for HFC-134a and CFC-12 is shown in Table 7 for both the case of similar mass fluxes and equivalent cooling (or heating) capacities. Unlike heat transfer enhancement factors, where high values of EF are desirable, low values of PF are desirable from the standpoint of using micro-fin tubes to increase refrigeration cycle performance. In this light, for evaporation the micro-fin tube performs better relative to the smooth tube with HFC-134a than it does with CFC-12. For HFC-134a, the PF is 1.3 compared to a CFC-12 value of 2.0 at the same mass flux and 1.8 at an equivalent cooling capacity. For condensation at low mass fluxes, HFC-134a has higher values of PF, indicating a potential reduction in performance. Specifically, the PF for HFC-134a is 1.8, while that of CFC-12 is about 1.3 to 1.4. At high mass fluxes during condensation and evaporation, the penalty factors are similar for the two refrigerants.

Significant increases in heat transfer while producing lesser increases in pressure drops has been the key advantage of the micro-fin tube over other in-tube enhancement techniques.

In this light, the relative magnitudes of enhancement factors and penalty factors compare similarly for HFC-134a and CFC-12 and the EF is greater than the PF, which is similar to the behavior observed in other refrigerants in the past. However, the one exception that should be

Table 7 Comparison of Pressure Drop Penalty Factors for HFC-134a and CFC-12

		Similar Mass Flux Comparison		**Similar Cooling (Heating) Capacity Comparison**	
		PF for HFC-134a	PF for CFC-12	PF for HFC-134a	PF for CFC-12*
Evaporation T = 10°C	low mass flux 150 kg/m²s	1.3	2.0	1.3	1.8
	high mass flux 300 kg/m²s	1.3	1.5	1.3	1.5
Condensation T = 40°C	low mass flux 150 kg/m²s	1.8	1.3	1.8	1.4
	high mass flux 300 kg/m²s	1.4	1.4	1.4	1.3

*Enhancement factors for CFC-12 are taken at mass fluxes that are increased by 30% (equivalent to the ratio of the enthalpy of vaporizations for HFC-134a and CFC-12).

noted is in the case of evaporation of HFC-134a, the penalty factor slightly exceeds the enhancement factor at the lowest temperature, 5°C, and the highest mass flux.

Additional investigations are required in order to explain this behavior.

CONCLUSION

Heat transfer and pressure drop data during evaporation and condensation of HFC-134a and CFC-12 were measured in a smooth tube and a micro-fin tube. For the smooth tube during evaporation, HFC-134a heat transfer coefficients were about 40% higher than they were for CFC-12, when compared at similar mass fluxes. During condensation, the HFC-134a heat transfer coefficients were about 25% higher. When HFC-134a and CFC-12 were compared at equivalent cooling (or heating) capacities, the HFC-134a heat transfer coefficients were about 10% higher for both evaporation and condensation. For the smooth tube, during both evaporation and condensation, pressure drops were greater for HFC-134a. However, when the two refrigerants were compared at the same cooling (or heating) capacity, the pressure drop of HFC-134a was less than that of CFC-12.

The performance of the micro-fin tube was compared to the performance of a smooth tube for HFC-134a and CFC-12 by forming heat transfer enhancement factors, EF, and pressure drop penalty factors, PF. Enhancement factors for HFC-134a varied from 1.5 to 2.5 during evaporation and from 1.75 to 2.5 during condensation. Enhancement factors for CFC-12 varied from 1.7 to 3.0 during evaporation and from 1.7 to 2.3 during condensation. In the case of CFC-12 and for most conditions with HFC-134a, penalty factors were less than enhancement factors. This behavior is similar to that observed in other refrigerants in the past. However, in the case of evaporation of HFC-134a, at the lowest temperature and highest mass flux, the penalty factor slightly exceeds the enhancement factor.

ACKNOWLEDGMENTS

Support for this project was provided by ASHRAE Research Project RP-630. The support and guidance provided by the Sponsoring Technical Committee, TC 1.3, Heat Transfer and Fluid Flow, and the Project Monitoring Committee (the members are Dr. Naim Azer, Dr. Floyd Hayes, Mr. Keith Starner, and Dr. Lynn Schlager) are greatly appreciated. Thanks are also extended to Dr. Petur Thors and the Wolverine Tube Company for supplying the micro-fin tubes used in this study.

NOMENCLATURE

A	= area
EF	= heat transfer enhancement factor
h	= convective heat transfer coefficient
PF	= pressure drop penalty factor
U	= overall heat transfer coefficient

Subscripts

i	= inner tube
o	= annulus
out	= outlet

REFERENCES

Eckels, S.J., and Pate, M.B. 1991. "In-tube evaporation and condensation of refrigerant-lubricant mixtures of HFC-134a and CFC-12." *ASHRAE Transactions*, Vol. 97, Part 2.

Khanpara, J.C., Pate, M.B., and Bergles, A.E., 1986. "Augmentation of R-113 in-tube condensation with micro-fin tubes." *Heat Transfer in Air Conditioning and Refrigeration Equipment*, **HTD-65**, pp. 21-32. New York: American Society for Mechanical Engineers.

Khanpara, J.C., Bergles, A.E., and Pate, M.B. 1987a. "A comparison of in-tube evaporation of R-113 in electrically heated and fluid heated smooth and inner-fin tubes." *Advances in Enhanced Heat Transfer-1987*, **HTD-68**, pp. 35-45. New York: American Society for Mechanical Engineers.

Khanpara, J.C., Pate, M.B., and Bergles, A.E., 1987b. "A comparison of local evaporation heat transfer enhancement for a micro-fin tube using refrigerants 22 and 113." *Boiling and Condensation in Heat Transfer Equipment*, **HTD-85**, pp. 31-39. New York: American Society for Mechanical Engineers.

Manwell, S.P., and Bergles, A.E. 1990. "Gas-liquid flow patterns in refrigerant-oil mixtures." *ASHRAE Transactions*, Vol. 96, Part2, pp. 456-464.

Reid, R.S., Pate, M.B., and Bergles, A.E., 1987. "Augmented in-tube evaporation of refrigerant 113." *Boiling and Condensation in Heat Transfer Equipment*, **HTD-85**, pp. 21-30. New York: American Society for Mechanical Engineers.

Schlager, L.M., Bergles, A.E., and Pate, M.B. 1987. "A survey of refrigerant heat transfer and pressure drop emphasizing oil effects and in-tube augmentation." *ASHRAE Transactions*, Vol. 93, Part 1, pp. 392-416.

Schlager, L.M., Pate, M.B., and Bergles, A.E. 1988. "Evaporation and condensation of refrigerant-oil mixtures in a smooth tube and micro-fin tube." *ASHRAE Transactions*, Vol. 93, Part 1, pp. 149-166.

Eckels, S.J. and M.B. Pate, *Evaporation and condensation of HFC-134a in a smooth tube and a micro-fin tube.* ASHRAE Transactions, 1991. **97**(2): p. 68-78.

Schlager, L.M., Pate, M.B., and Bergles, A.E. 1989a. "Heat transfer and pressure drop during evaporation and condensation of R-22 in horizontal micro-fin tubes." *International Journal of Refrigeration*, Vol. 12, No. 1, pp. 6-14.

Schlager, L.M., Pate, M.B., and Bergles, A.E. 1989b. "Evaporation and condensation heat transfer and pressure drop in horizontal, 12.7 mm micro-fin tubes with refrigerant 22." *Heat Transfer Fundamentals, Design, Applications, and Operating Problems*, **HTD-108**, pp. 205-213. New York: American Society for Mechanical Engineers.

Schlager, L.M., Pate, M.B., and Bergles, A.E. 1989c. "Performance of micro-fin tubes with refrigerant-22 and oil mixtures." *ASHRAE Journal*, November, pp. 17-28.

Schlager, L.M., Pate, M.B., and Bergles, A.E. 1989d. "A comparison of 150 and 300 SUS oil effects on refrigerant evaporation and condensation in a smooth tube and a micro-fin tube." *ASHRAE Transactions*, Vol. 95, Part 1, pp. 387-397.

Schlager, L.M., Pate, M.B., and Bergles, A.E. 1989e. "Heat transfer and pressure drop performance of smooth and internally finned tubes with oil and refrigerant 22 mixtures." *ASHRAE Transactions*, Vol 95, Part 2.

Schlager, L.M., Pate, M.B., and Bergles, A.E. 1990a. "Performance predictions of refrigerant-oil mixtures in smooth and internally finned tubes-Part I: Literature review." *ASHRAE Transactions*, Vol. 96.

Schlager, L.M., Pate, M.B., and Bergles, A.E. 1990b. "Performance predictions of refrigerant-oil mixtures in smooth and internally finned tubes-Part II: Design equations." *ASHRAE Transactions* Vol. 96.