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The effect of oil on the two-phase critical flow of Refrigerant 134a through short tube orifices

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Abstract—An experimental investigation of two-phase flow of mixtures of Refrigerant 134a with 168 SUS polyalkylene glycol (PAG) oil through short tube orifices was performed for oil concentrations ranging from 0 to 5.1%. Both two-phase and subcooled liquid flow entering short tubes were studied for an upstream pressure of 1172 kPa, for subcoolings as high as 13.9°C, and for qualities as high as 8% at the inlet of the short tube. Downstream pressures were varied from saturation pressure, P_{sat}, to 310 kPa. The effects of the lubricant on the flow characteristics were discussed as a function of downstream pressure, upstream subcooling/quality and upstream pressure. The effects of oil concentration on mass flow through short tubes varied as a function of upstream subcooling or quality. The maximum reduction in flow occurred at zero subcooling, where the flow was reduced by approximately 12.1% for a 5.1% oil concentration. For entering subcooling above 8.3°C and at a quality of 5%, there were small increases in flow for the addition of oil to the refrigerant. The observed flow trends were analyzed using pressure profile measurements and visualization tests.

INTRODUCTION

LUBRICANTS are required in all vapor compression refrigeration systems. The lubricant circulated with the refrigerant can affect the flow rate through short tube orifices that are used as expansion devices in heat pumps or air conditioners. The effect of lubricants on the two-phase heat transfer during evaporation and condensation has been well documented in recent years [1–3]. However, no studies have focused on the effect of the lubricant on the two-phase critical flow through expansion devices such as capillary tubes and short tube orifices.

The flow of refrigerants through short tube orifices was investigated by several previous researchers [4–9]. However, their focus was on pure CFC-12 or HCFC-22 for subcooled and saturated liquid entering the short tubes. Recently, due to a wide applications of short tube orifices in heat pumps and air conditioners, Mei [6], Aaron and Domanski [8], and Kuehl and Goldschmidt [9] studied the flow of HCFC-22 through short tubes and developed flow rate prediction models. The designs for their experimental setups were based on the refrigeration cycle, which included a compressor. Their data were taken for uncontrolled and untested oil concentrations.

In this paper, the results of an experimental study with HFC-134a/lubricant mixtures flowing through short tube orifices with oil concentrations of 0, 2.1 and 5.1% were reported for variations in downstream pressure, upstream subcooling/quality, and upstream pressure. The lubricant used was 168 SUS polyalkylene glycol (PAG) at 38°C. The observed flow trends were analyzed using the pressure profile measurement tests and visualization tests.

EXPERIMENTAL APPARATUS

Experimental setup

A schematic diagram of the experimental setup is shown in Fig. 1. The test setup used was capable of measuring the effect of operating variables on the mass flow rate and flow characteristics. The system was designed to allow easy individual control of operating variables such as upstream subcooling or quality, upstream pressure, and downstream pressure. It also allowed for changing the oil concentration by injection of oil into the system. The test rig consisted of three major flow loops: (1) a refrigerant flow loop containing a detachable test section, (2) a hot water flow loop used for an evaporation heat exchanger and (3) a water–glycol flow loop used for a condensation heat exchanger.

A diaphragm liquid pump with a variable speed motor was used to provide a wide range of refrigerant mass flow rates. The pump did not require any external lubrication. Oil concentration was an adjustable variable in the system. The pressure entering the test section (upstream or condenser pressure) was controlled by adjusting the speed of the refrigerant pump. A hand-operated needle valve was utilized to permit precise control of upstream pressure by bypassing liquid refrigerant from the pump to the short tube exit. The refrigerant flow rate was measured by a turbine flow meter in the liquid line between the pump and the evaporation heat exchanger.

NOMENCLATURE

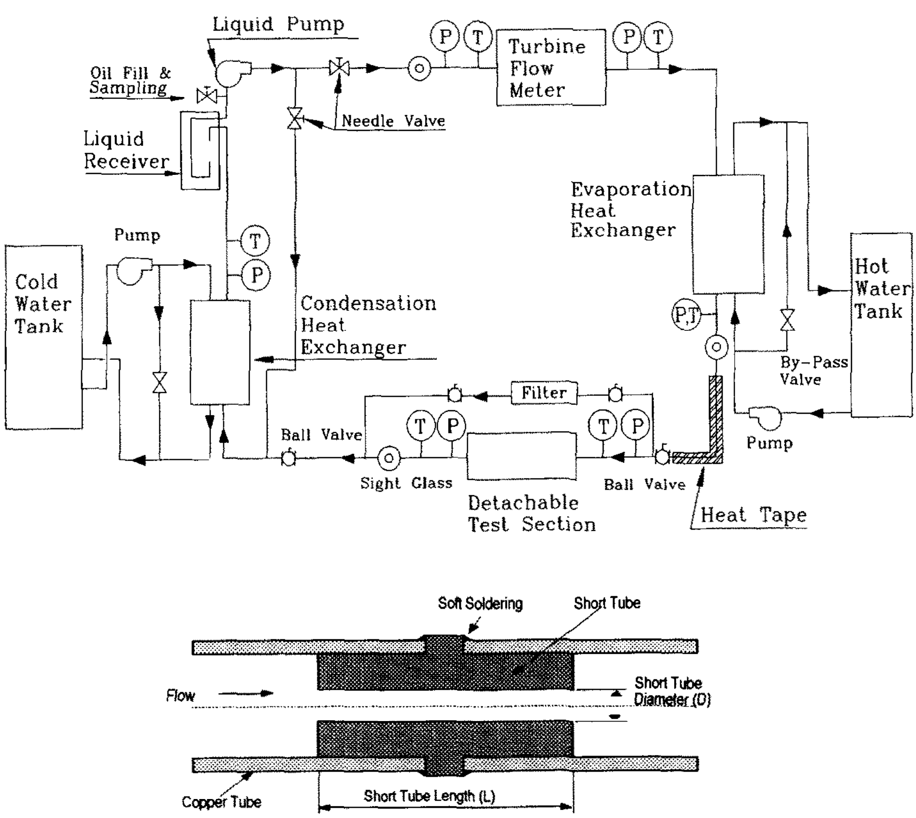
C_p specific heat at constant pressure [kJ kg^{-1} °C^{-1}]
D short tube diameter [mm]
L short tube length [mm]
$\dot{m}$ mass flow rate [kg h^{-1}]
m_R mass flow ratio of the oil/refrigerant mixture to pure refrigerant
P pressure [kPa]
T temperature [°C]
W mass fraction of refrigerant to the oil/refrigerant mixture.

Greek symbol
ρ density.

Subscripts
down downstream of orifice
m oil–refrigerant mixture
o oil
r refrigerant
sat saturated conditions upstream of orifice
up upstream of orifice.

The refrigerant subcooling or quality entering the test section was set by a water-to-refrigerant heat exchanger (evaporation heat exchanger) and a heat tape. For single-phase conditions at the inlet of the test section, most of the energy transfer to the refrigerant was supplied by the evaporation heat exchanger. A heat tape with adjustable output from 0 to 0.9 kW was utilized to provide precise control of upstream subcooling. For two-phase flow conditions at the inlet of the test section, the flow from the pump was heated by the evaporation heat exchanger to 1.1°C of subcooling, and a heat tape was used to reheat the refrigerant to the desired level of quality. For two-phase conditions upstream of the orifice, power input into a heat tape was measured using a watt transducer. Liquid refrigerant temperature entering the heat tape and the inside and outside temperatures of the insulation were also measured to calculate heat loss through the insulation of the heat tape. Inlet quality was calculated from an energy balance between the heat tape and the short tube orifice.

The pressure and temperature were measured upstream and downstream of the short tube. Two-phase refrigerant exiting the test section was con-



Detachable Test Section

FIG. 1. Schematic of the short tube test setup.

Table 1. Dimensions of the test sections tested in the present study

Test measurement	Length (mm)	Diameter (mm)
Routine flow	12.70	1.34
Pressure profile	12.83	1.33
Visualization	12.70	1.27

densed and subcooled in the water–glycol condensation heat exchanger so that the refrigerant pump had only liquid at its suction side. The pressure at the exit of the test section (downstream or evaporator pressure) was controlled by adjusting the temperature and flow rate of chilled water–glycol entering the heat exchanger. The water–glycol loop consisted of a 757 l insulated storage tank, 17.5 kW chiller unit, a centrifugal pump, and an in-line heater.

Three separate short tubes were used for the experiments (Table 1). There included an orifice for routine mass flow tests, an orifice for pressure profile measurement tests and a transparent orifice for visualization tests. The short tube used in the routine flow tests was made from brass, and then fixed between two 9.53 mm diameter copper tubes with soft solder. A study for pressure profile measurements were performed using a specially designed short tube. Five pressure taps were bored to 0.20 mm diameter inside the tube, and two pressure taps were bored to 0.51 mm diameter before and after the tube. The effects of pressure taps on flowrate were examined by comparing the difference of mass flow rate between a short tube without pressure tap and a short tube with pressure taps. The differences were less than $\pm 4\%$. A glass short tube was manufactured to allow visual study of the flow. The glass short tube section was fixed between two 9.53 mm diameter glass tubes by flame solder. A detailed description for the tested short tubes can be found in ref. [10].

Diameters were measured using a precise plug gauge set with 0.013 mm increment of diameter. Short tube lengths were measured with a dial caliper. The estimated accuracy of both diameter and length measurements were ± 0.013 mm. Experimental uncertainties were estimated as $\pm 0.2°C$ for temperature, $\pm 0.2\%$ of full scale (3447 kPa) for pressure, $\pm 0.5\%$ of full scale (0.1 l s^{-1}) for volumetric flow rate and $\pm 0.5\%$ full scale (1.5 kW) for watt transducer. Refrigerant mass flow rate was calculated from measured volumetric flow rate, pressure and temperature across the flow meter. The two-phase quality at the inlet of the short tube was determined by applying an energy balance to the heat tape at the entrance of the short tube. The experimental uncertainties of mass flowrates and qualities were estimated using the Kline and McClintock [11] error method. Based on sample calculations, the uncertainties of the mass flowrates and the qualities were less than 3% of calculated mass flowrates and 3.5% of calculated qualities, respectively.

Oil injection and sampling

The lubricant was injected into the suction side of the refrigerant pump using an air-cylinder in a batch process. The testing sequence proceeded from pure refrigerant to progressively higher oil concentrations. The amount of the lubricant injected was calculated from the rod displacement and the diameter of the cylinder. Two oil concentrations were used in this study: 2.1 and 5.1%. To achieve these concentrations, a total of 229 and 555 g were added to the system. During injection into the system, the weight of the lubricant was measured using an electronic scale that was accurate to ± 13.6 g.

Oil concentration was determined by sampling. The sampling procedure and calculation of the oil concentration was based on ASHRAE Standard 41-4-1984 [12]. The volume of the sampling vessel (12.7 cm I D $\times$ 30.5 cm long) was large compared to the volume of the sample (0.454 kg $\pm$ 10% of the sample) to ensure low vapor velocity during the distilling procedure so that no oil particles could leave with the vapor. After sampling, the refrigerant was removed from the sampling vessel by slowly bleeding the refrigerant vapor through a bleeder assembly which included filter and a capillary tube to catch any entrained oil in the exiting refrigerant. After bleeding, the cylinder was evacuated to remove any dissolved refrigerant in the lubricant. Based on the measurement of the empty weight of sampling unit, the weight after sampling, and the weight after bleeding off the refrigerant, the oil concentration in the refrigerant was calculated. Because of the small weight of the oil in the samples (typically less than 25 g), a different and more accurate scale was required from the one use to measure the total weight of the oil injected in the system. This second scale was accurate to within ± 0.5 g. The estimated accuracy of the weighing scale was $\pm 0.1\%$ of sampled refrigerant–lubricant mixtures.

Test conditions and procedure

A series of measurements for oil concentrations of 0, 2.1 and 5.1% was run to investigate the influence of lubricant on flow rate as a function of operating parameters and short tube geometry. Experimental conditions were chosen to cover a wide range of operating conditions for a short tube expansion device found in a typical residential heat pump or air-conditioner. Upstream pressure was set at 896, 1172, and 1448 kPa, while downstream pressure was varied at 310, 379, and 483 kPa. For single-phase entering the short tube, the subcooling was varied between 0 and 13.9°C. For the two-phase case, quality was set over range from 0 to 8%.

When the HFC-134a tests were completed, the 168 SUS PAG was injected into the suction side of the pump. Before sampling of the refrigerant–lubricant mixture to measure oil concentration, the system was run for 3 h to allow the refrigerant and lubricant to fully mix. After completing the series of the tests for a certain oil concentration, the oil concentration was increased by injecting more lubricant and then the same series of the tests was repeated.

The data developed from the measurements included refrigerant flow rate, pressure drop across the short tube, upstream subcooling/quality, and pressure distribution along the short tube. Steady state data were collected every five seconds and averaged for a period of four minutes. The modified Benedict–Webb–Rubin (MBWR) equation of state was used for calculation of HFC-134a properties [13].

The density of the refrigerant–lubricant mixture was used to convert the measured volumetric flow rate to mass flow rate. For the pure refrigerant, the density of liquid flow was directly determined from the measured temperature and pressure before and after the flow meter. The density of the refrigerant–lubricant mixtures was adjusted for oil concentrations by using an ideal mixing equation [14] given by:

$$\rho_m = \frac{\rho_L}{1+W(\rho_L/\rho_r-1)} \tag{1}$$

where ρ_m is the density of mixture and W is the mass fraction of refrigerant.

The specific heat of the refrigerant–lubricant mixtures, C_{p_m}, was calculated by adjusting the specific heat for pure refrigerant and lubricant [15]:

$$C_{p_m} = C_{p_r}W + C_{p_L}(1-W). \tag{2}$$

The enthalpy of vaporization and the temperature–pressure relationship at saturation were assumed to be unaffected by the existence of the lubricant.

EXPERIMENTAL RESULTS AND DISCUSSION

Effects of downstream pressure

The establishment of choking conditions for different oil concentrations was studied by the comparison of the mass flow rate change and pressure profile for a pure refrigerant and the refrigerant–lubricant mixtures. Figure 2 shows the mass flow rate as a function of downstream pressure for different oil concentrations and for a constant upstream subcooling of 13.9°C. As the oil concentration increased, the mass flow rate increased for a given downstream pressure. For all three concentrations, there was a small increase (approximately 7%) in flow rate as the downstream pressure was decreased from near saturation (780 kPa) down to as low as 310 kPa. This small increase in mass flow has been noticed by previous investigators in nozzles with saturated water [16] and short tube orifices with refrigerant 22 [8]. Silver and Mitchell [16] hypothesized that the increase in flow rate was caused by increased heat loss from the nozzle as the downstream pressure was decreased. Because the downstream condition is saturated, a decrease in pressure results in a decreasing downstream temperature.

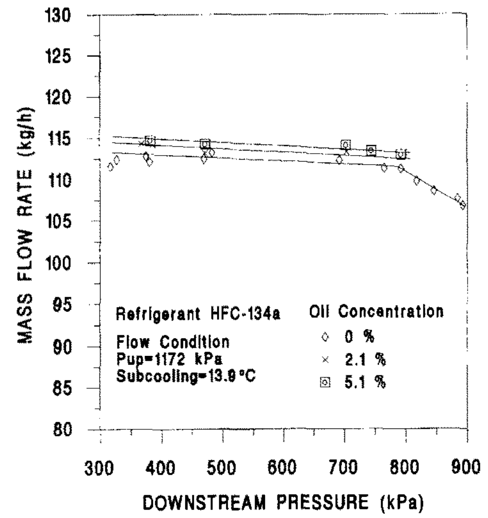


FIG. 2. Effects of oil concentration on choking phenomena for the flow through a short tube with $L = 12.70$ m and $D = 1.34$ mm.

For instance, the decrease in downstream pressure from 780 to 310 kPa resulted in a drop in downstream temperature of 29°C. For two-phase flow through orifices, the fact that the mass flow rate showed little change with respect to downstream pressure below 800 kPa for all three oil concentrations would indicate that the flow rate was nearly choked.

The existence of choked (or nearly choked) flow for mixtures can be verified using the results of pressure profile measurements. Figure 3 shows the effects of downstream pressure on the pressure profile throughout the short tube for the pure refrigerant. The pressure dip at the first pressure tap would indicate the presence of a vena contracta near the entrance of the orifice. For the downstream pressures shown in Fig. 3, the pressure at the first pressure tap ranged from

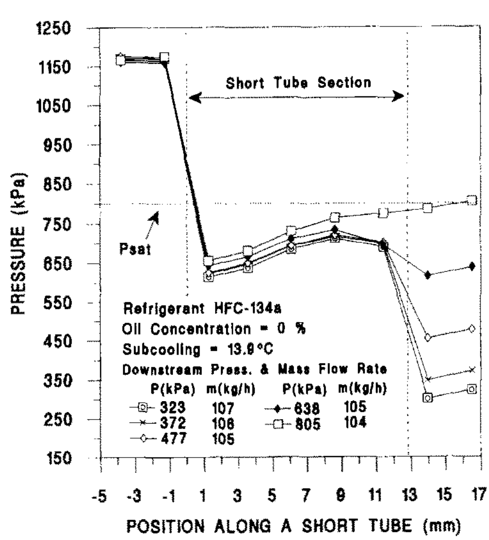


FIG. 3. Pressure profile for pure HFC-134a along a short tube with $L = 12.83$ mm and $D = 1.33$ mm as a function of downstream pressure.

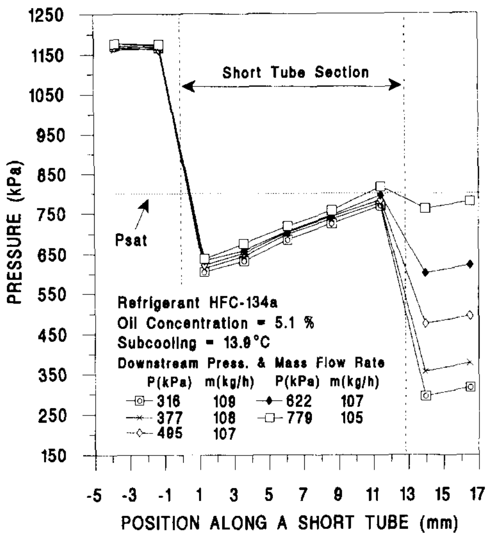


FIG. 4. Pressure profile for oil concentration of 5.1% along a short tube with $L = 12.83$ mm and $D = 1.33$ mm as a function of downstream pressure.

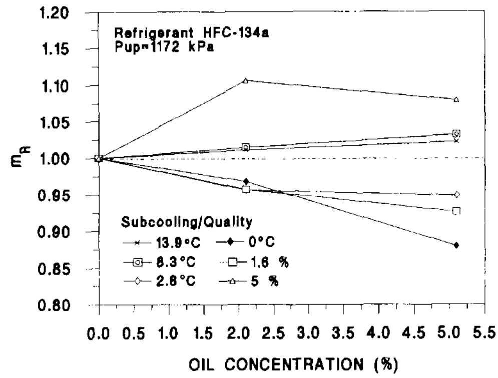


FIG. 5. Mass flow ratio showing the effects of oil concentration for a short tube with $L = 12.70$ mm and $D = 1.34$ mm.

150 to 200 kPa less than the saturation pressure corresponding to the entrance temperature. From the visualization studies, no vaporization occurred near the first pressure tap for any of the conditions in Fig. 3. Thus, the fluid was 'underpressurized', or superheated, at the first pressure tap and in a metastable state. Vaporization only began to occur toward the exit of the orifice as indicated by the drop in pressure by the pressure tap located at 11.4 mm inside the orifice in Fig. 3.

The entrance pressure drop for a given downstream pressure was within 1% for all oil concentrations tested at $P_{down} < P_{sat}$. The pressure profiles for different downstream pressures at an oil concentration of 2.1% were similar to that for pure HFC-134a. However, as the oil concentration increased to 5.1%, the pressures after the first pressure tap appeared to recover linearly to near P_{sat} at the last pressure tap (Fig. 4). The pressure at the last pressure tap was over 75 kPa higher for the 5.1% oil concentration compared to the pure refrigerant case. This trend would suggest that the increase of the oil concentration extended the pressure recovery region and caused a higher maximum pressure inside the tube. The increasing pressure inside the orifice after the first pressure tap would also suggest that the presence of the lubricant may have delayed the onset of flashing until the outlet of the orifice. Manufacturer's data of the lubricant indicated that it was soluble with HFC-134a for the liquid conditions at the entrance of the tube in Figs. 3 and 4. The presence of the oil would be expected to provide a small increase in the boiling temperature of the refrigerant. This effect may be responsible for the delayed flashing. If vaporization were occurring in the orifice near the last two pressure taps, a pressure drop would be expected from the acceleration of the fluid as noted in the pure case in Fig. 3. However, none is present in Fig. 4. Even though the pressure profile for the highest oil concentration (5.1%) was different from that for a pure refrigerant, the pressures throughout the short tube showed only a small dependence on downstream pressures ($P_{down} < P_{sat}$) which would indicate that the flow was nearly choked.

Effects of upstream subcooling/quality

One method that can be used to quantify the effects of oil concentration on mass flow rate as a function of upstream subcooling or quality is to define a new term, the mass flow ratio, m_R, as:

$$m_R = \frac{\text{mass flow rate of the oil–refrigerant mixture}}{\text{mass flow rate of pure refrigerant}} = \frac{\dot{m}_{mix}(P_{up}, P_{down}, T_{up}, L, D)}{\dot{m}_{pure}(P_{up}, P_{down}, T_{up}, L, D)} \quad (3)$$

where P_{up} and P_{down} are upstream pressure and downstream pressure, respectively. T_{up} indicates a value of upstream subcooling or quality. The definition of m_R allows direct comparison of the flow rate with a lubricant to the flow rate without the lubricant for a given set of conditions. If the oil increases the flow rate, then m_R will be greater than one. Any degradation of flow will result in a value of m_R less than one.

Figure 5 shows a plot of mass flow ratio versus oil concentration for different upstream subcoolings and qualities in a short tube 12.70 mm long and 1.34 mm in diameter. The effects of lubricant on flow rate varied as a function of upstream subcooling or quality. The value of m_R was slightly greater than one at high subcooling (> 8.3°C), decreased until reaching a minimum at 0°C subcooling, then increased as the quality increased. For subcooling greater than or equal to 8.3°C, the oil had a very small (less than 3.2%), but positive effect on the flow rate. The lubricant had the largest effect on the flow when the state of the refrigerant was a saturated liquid (0°C subcooling) at the entrance. The flow dropped by 12.1% for a 5.1% oil concentration and entrance conditions at 0°C subcooling. The maximum flow enhancement (10.6%)

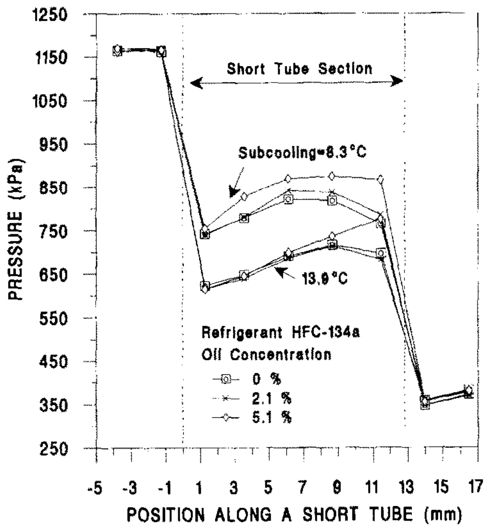


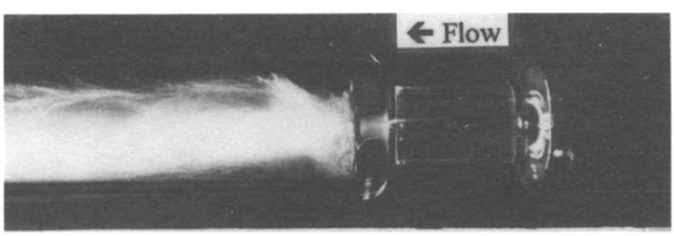
FIG. 6. Pressure profile along a short tube with $L = 12.70$ mm and $D = 1.34$ mm for subcooling of 13.9 and 8.3°C.

occurred for entrance conditions at 5% quality and an oil concentration of 2.1%. For subcooling between 0°C and 8.3°C, the slope of m_R decreased with an increase of subcooling.

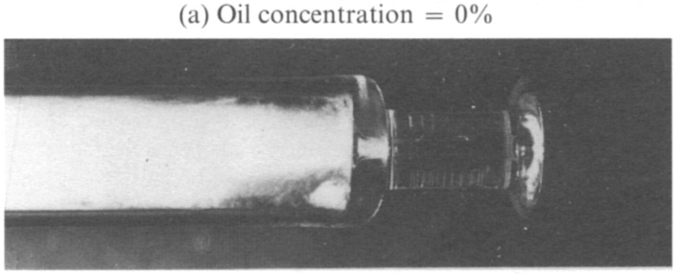
The diversity of effects of oil on the flow rate was unexpected. One might have expected the presence of a lubricant to either have uniformly decreased or increased all flows through the orifice. The pressure distribution data provide some insight on potential explanations of the flow behavior seen in Fig. 5.

First, at high levels of subcooling (greater than or equal to 8.3°C), the oil had little impact on the pressure dip near the entrance, but increased the pressure near the outlet of the orifice (Fig. 6). This trend became more pronounced as the oil concentration increased. The difference between the pressure at the last pressure tap for oil concentrations of 0 and 5.1% was 90 kPa for 13.9°C subcooling. Increasing the pressure toward the end of the tube would delay the onset of flashing which should provide for a higher flow rate (note the differences in the location of onset of flashing for the pure refrigerant and 5.1% oil concentration for 8.3°C subcooling in Figs. 7(a) and (b). The change of location of the initiation of flashing due to the presence of the oil had a small effect (less than 3.2%) on the flowrate. The effect of flashing point movement on mass flow was typically small because even for the pure refrigerant case, flashing occurred only toward the exit of the orifice. Thus, it would appear that a larger metastable region existed for the higher oil concentrations than with pure HFC-134a. As discussed in the previous section, the extension of the metastable region (delay in flashing) could be caused by the oil's solubility in the refrigerant. Thus, for high subcooling at the entrance, the enhancement of the flow rate might result from the slight movement of the flashing point because of the presence of oil.

The second effect of the oil was that it decreased the entrance pressure drop when the entering refrigerant was near saturated liquid conditions. The pressure drop at the first pressure tap was observed in the subcooling region regardless of oil concentration and in the saturation region for most oil concentrations. Figures 8 and 9 show the pressure distribution for 2.8° and 0°C subcooling at the inlet, respectively, for 0, 2.1 and 5.1% oil concentrations. The differences between these two figures and Fig. 6 is substantial. Higher oil concentrations produced a significant change in the inlet pressure drop for entering refrigerant conditions near saturated liquid. For example, the pressure drop at the first pressure tap for 2.5°C subcooling was 332 kPa for the pure refrigerant case, but decreased to 294 kPa for 2.1% lubricant concentration, and 226 kPa for 5.1% oil concentration. In contrast, for 13.9°C



(a) Oil concentration = 0%



(b) Oil concentration = 5.1%

FIG. 7. Photographs for oil concentrations of 0 and 5.1% at a upstream subcooling of 8.3°C.

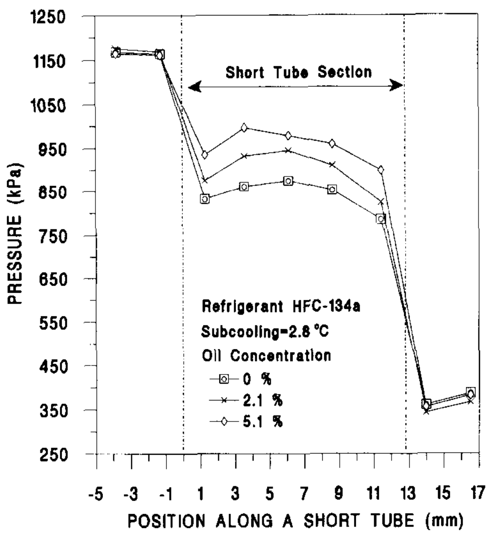


FIG. 8. Pressure profile along a short tube with $L = 12.83$ mm and $D = 1.33$ mm for a upstream subcooling of 2.8°C.

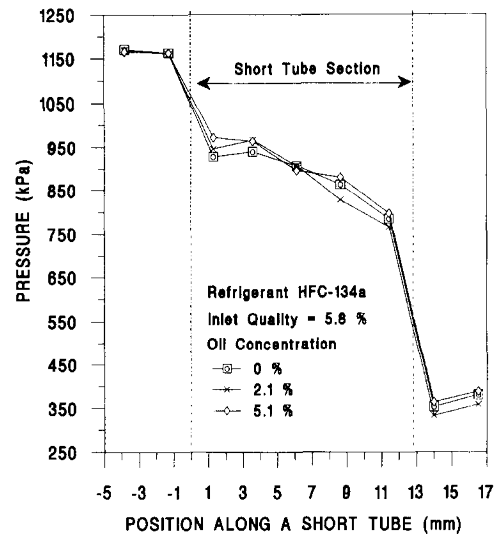


FIG. 10. Pressure profile along a short tube with $L = 12.83$ mm and $D = 1.33$ mm for inlet quality of 5.8%.

subcooling, the pressure drop at the first tap with an oil concentration of 5.1% was 9 kPa higher than with the pure refrigerant. For 8.3°C subcooling, the entrance pressure drop with a 5.1% oil concentration was within 8.5 kPa of that for the pure refrigerant case.

Related to the second effect was that the oil appeared to raise the pressure of the mixture throughout the orifice as the oil concentration increased for inlet conditions near saturated liquid. For both the 2.8 and 0°C subcooling cases (Figs. 8 and 9), the pressure at each location inside the orifice increased with increasing concentrations of the lubricant. This behavior was different from the high subcooling cases where the pressure increased only toward the exit of the orifice (Fig. 6).

For near saturation conditions at the inlet, the point

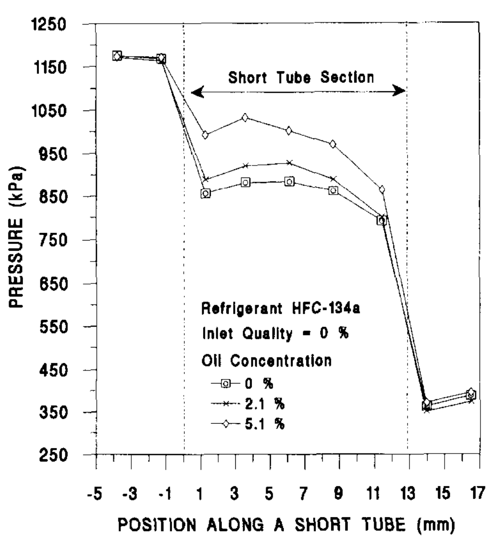
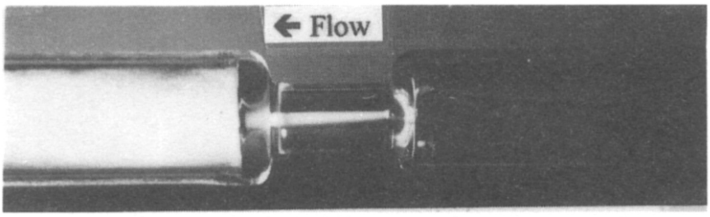


FIG. 9. Pressure profile along a short tube with $L = 12.83$ mm and $D = 1.33$ mm for a inlet quality of 0%.

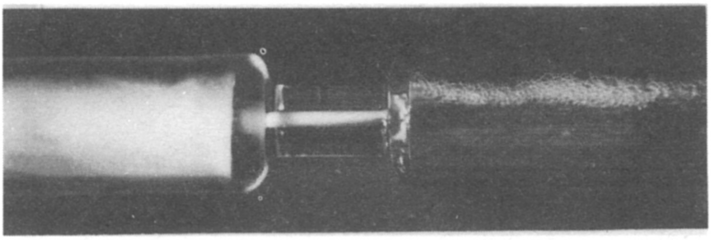
at which the pressure began to decrease was also shifted toward the entrance of the orifice as the oil concentration increased. For example, with 0°C subcooling, the pressure peaked at the approximately 6.1 mm inside the orifice for the pure refrigerant case. For 5.1% oil concentration, the peak was at approximately 3.6 mm inside the short tube. The shift in peak pressure would suggest that initiation of flashing was shifted toward the entrance of the short tube as the oil concentration was increased. Shifting the initiation of flashing toward the entrance should reduce the amount of mass flow through the orifice because more vapor would be present at the orifice exit when the flow was choked.

For the highest entrance quality case (5.8%), there was a relatively large enhancement in flow. The exact reason for this enhancement was not clear. The pressure drop at the first pressure tap showed only a small decrease as the oil concentration increased (Fig. 10). However, the pressure distributions for the highest entrance quality case throughout the orifice were similar regardless of oil concentration. With similar pressure distributions (Fig. 10), the presence of the oil in the two-phase flow could have decreased the void fraction of the fluid inside the orifice because the oil did not vaporize. However, no measurements were made of the void fraction.

Another difference at higher oil concentrations and higher inlet qualities was the presence of foaming of the mixture at the interface of liquid and vapor (Fig. 11). Because the oil has a higher surface tension than the refrigerant, increasing the oil concentration would also be expected to increase the surface tension of the oil/refrigerant mixture. Nucleation theory would indicate that increasing the surface tension of the mixture would increase the amount of superheat needed in the liquid for the initiation of vaporization. Thus, the increased surface tension due to the presence of



(a) Oil concentration = 0%



(b) Oil concentration = 5.1%

FIG. 11. Photographs for oil concentrations of 0 and 5.1% at a inlet quality of 5%.

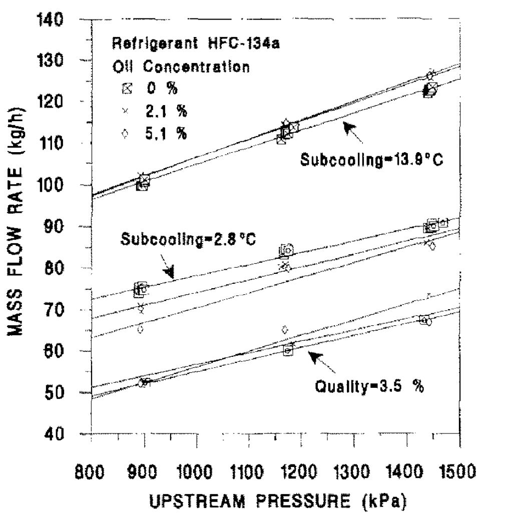


FIG. 12. Effects of oil concentration on mass flow rate as a function of upstream pressure for a short tube with $L = 12.70$ mm and $D = 1.34$ mm.

the oil should delay vaporization in the short-tube compared to the case where no oil is present in the refrigerant [17]. Delaying nucleation would be expected to help increase the flow through the short tube. No data were collected to confirm this hypothesis.

CONCLUSIONS AND RECOMMENDATIONS

The effect of oil on two-phase critical flow of a refrigerant through short tube orifices appears to be a complex process. The results indicated that oil provided small enhancements to the flow for high subcooling conditions and at the highest quality (5.8%) used for this study. For entrance conditions near the saturation region, the oil created a drop in the flow through the orifice. Some of the increased or decreased flow was accompanied by a measurable change in the pressure distributions in the orifice as well as changes in the visual location of where flashing was initiated.

This study focused on one refrigerant (HFC-134a) and one lubricant (168 SUS polyalkylene glycol). It would be difficult to generalize from this study that other combinations of refrigerants and oils will produce similar effects. It would appear that there is a need to better understand the physical mechanisms involved in how the oil affects the flow through the orifices. One recommendation would be to investigate different combinations of refrigerants and oils to identify and quantify relevant physical variables that affect the flow with lubricants.

This study was also limited to one geometry (orifice length and diameter). If one of the effects of the oil is to reduce the losses at the entrance to the tube, the size of this effect may be dramatically different for a different sized diameter or length orifice.

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