A Comparison of the Effects of Different Lubricants on the In-Tube Evaporation of an HFC-Blend Refrigerant

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ABSTRACT

Heat transfer coefficients are an important consideration when selecting a refrigerant, however, heat transfer is affected by the type and amount of lubricant circulation. HFC-blend R-404A is a leading candidate to replace R-502. The focus of this study was a comparison of the effects that different lubricants, such as a naphthenic mineral oil, a polyolester, and an alkylbenzene have on the in-tube evaporation of R-404A. In-tube evaporation heat transfer coefficient measurements were made in both a smooth tube and a micro-fin tube. The results support the contention that polyolester lubricants exhibit superior heat transfer characterization in relation to other hydrocarbon lubricants.

INTRODUCTION

Refrigerant mixtures and blends are being considered as potential replacements for R-502 refrigeration systems, however, the heat transfer characteristics of any replacement refrigerant must be known before the final selection is made. In addition to the in-tube evaporation heat transfer performance of the refrigerant, the effect on heat transfer of the type and amount of lubricant mixed with the circulating refrigerant must also be determined. In fact, the performance of the lubricant mixed with the refrigerant in the evaporator will be an important factor in the decision as to what type of lubricant to use in a refrigeration system [Sundaresan, 1992 and 1993].

In this study a commercially available HFC-blend, R-404A (R-125(42%)/R-143a(52%)/R-134a(4%)) was tested during in-tube evaporation at 0°C in both a smooth tube and a micro-fin tube. The refrigerant was tested with lubricant mass concentrations of 1 and 2% for a mineral oil, an alkylbenzene, and a polyolester lubricant (POE). These refrigerant-lubricant mixture results were compared to a baseline of pure R-404A results.

The facility that was used for these tests has been used in the past for various evaporation and condensation tests of both refrigerant mixtures and pure refrigerants, such as R-12, R-113, R-22, and R-134a [Eckels, et al. 1994, Doerr, et al. 1994]. These past tests were performed with and without lubricants for both smooth and micro-fin tubes.

TEST FACILITY

The test facility used in this study consists of four main parts: a refrigerant loop, a water loop, a data acquisition system, and a dual test section. The following sections provides detailed descriptions of the four main parts of the test facility. A schematic of the test rig is shown in Figure 1.

Test Section

The test sections consist of horizontal test tubes (a smooth tube and a micro-fin tube) and a surrounding annuli. One tube is a 3/8" O.D. copper tube which is 3.67 m long while the other is a 3/8" O.D. micro-fin tube which is also 3.67 m long. The annulus which surrounds the tube is also 3.67 m long and is constructed of a copper tube with a 17.2 mm inside diameter. The test tubes are centered in the annuli by a series of spacers. The spacers are constructed of three stainless steel rods spaced 120 degrees about the annulus.

The test sections are instrumented with temperature and pressure sensors. The temperatures are measured with resistance type temperature probes, RTDs, which have been calibrated to an accuracy of ±0.05°C. The pressure is measured with a strain-gage type pressure transducer which is accurate to ±9 kPa. The pressure drop in the test section is measured with a strain-gage type differential pressure transducer accurate to ±0.2 kPa.

Refrigerant Loop

The refrigerant loop consists of an after-condenser, a positive displacement pump, an accumulator, and a boiler. The after-condenser is a co-axial heat exchanger which condenses and subcools the refrigerant leaving the test section. The water-glycol mixture for the heat exchanger is provided by a R-502 chilling unit. After being subcooled, the refrigerant is circulated by a positive displacement pump. The pressure in the test section is controlled by the bladder accumulator. This accumulator also helps to dampen out pressure fluctuations that may occur in the system. The quality of the refrigerant entering the test section is set by a heater located directly upstream of the test section. The heater is a 12.7 mm O.D. by 2.63 m long stainless-steel tube heated by direct electrical current. The heater is electrically isolated from the rest of the system by high-pressure rubber hoses. The refrigerant mass flow rate is measured by a coriolis-type mass flow meter accurate to 0.15% of the mass flow rate plus 2.25e-05 kg/s.

Water Loop

The water loop consists of a centrifugal pump, an in-line electric heater, and a heat exchanger. The mass flow rate is controlled by a valve that restricts the flow of water. The temperature of the water entering the test section is controlled by both the electric heater and the heat exchanger, which uses building supplied chilled water to cool the water in the test loop. The water mass flow rate is also measured by a coriolis type flow meter with an accuracy of 0.15% of the mass flow rate plus 2.25e-04 kg/s.

Data Acquisition

Data acquisition is done with a personal computer, a 40 channel scanner, and a multimeter. The controlling program on the personal computer is written in FORTRAN and controls the multimeter and the scanner via a IEEE-488 bus.

EXPERIMENTAL PROCEDURE

The test facility is allowed to come to steady state before the final data acquisition is done. This is achieved by setting the mass flow rates, the refrigerant quality, and the annulus water temperatures, with the latter controlling the outlet quality. The data acquisition system then scans for temperature, mass flow rate, and pressure fluctuations. When the fluctuations are minimal the final data acquisition program is run. Each of the channels is scanned a total of five times while the pressure is scanned 35 times because of pressure drop fluctuations. The inlet quality is maintained between 8 and 15% while the outlet quality is kept between 80 and 85%.

DATA ANALYSES

Raw data from the data acquisition system are analyzed for each run to determine the in-tube heat transfer coefficient and the quality. The main equations used in processing the raw data are based on energy balances. The energy transferred in the test section is computed from an energy balance on the water side.

$$Q_w = M_w \cdot Cp_w \cdot (T_{w,out} - T_{w,in}) \tag{1}$$

The vapor quality entering the test section is calculated from an energy balance on the refrigerant boiler. The heat output of the boiler Q_h is calculated from voltage and current readings. The heat input to the refrigerant takes two forms, sensible and latent.

$$Q_{sens} = M_r \cdot Cp_r \cdot (T_{r,sat} - T_{h,in}) \tag{2}$$

$$Q_{lat} = M_r \cdot h_{fg} \cdot X_{h,out} \tag{3}$$

The saturation temperature of the refrigerant entering the boiler is determined from the pressure of the refrigerant entering the test section. Equation 2 can then be used to determine the quality of the refrigerant entering the test section.

$$X_{in} = (1/h_{fg}) \cdot [\, (Q_h / M_r - Cp_r \cdot (T_{sat} - T_{h,in})\,] \tag{4}$$

The quality change of the refrigerant in the test section is determined from an energy balance on the water side

$$\Delta X = Q_w / (M_r \cdot h_{fg}) \tag{5}$$

The refrigerant-side heat transfer coefficient is determined from an overall heat transfer coefficient and the annulus-side heat transfer coefficient. The annulus-side heat transfer coefficient, h_o, was determined by using a modified Wilson plot technique over the range of flow rates and temperatures encountered during evaporation tests. The correlation for the annulus-side heat transfer coefficient is a Dittus-Boelter type equation. The overall heat transfer coefficient is determined from the energy balance on the test section.

$$Uo = Qw / (Ao \cdot LMTD) \tag{6}$$

The log mean temperature difference is determined from the inlet and outlet temperatures on the water and refrigerant sides.

$$LMTD = (\Delta T_1 - \Delta T_2) / \ln (\Delta T_1 / \Delta T_2) \tag{7}$$

where

$$\Delta T_1 = T_{r,out} - T_{w,in} \tag{8}$$

$$\Delta T_2 = T_{r,in} - T_{w,out} \tag{9}$$

Assuming the thermal resistance of the copper tubing as negligible, the refrigerant-side heat transfer coefficient is then determined from

$$h_i = 1 / (1/U_o - 1/h_o) A_i / A_o \tag{10}$$

The heat transfer coefficient determined in this equation is an average value over the length of the tube.

RESULTS

Data are presented and comparisons are made for several different lubricants, namely, a POE, a mineral oil, and an alkylbenzene, mixed with an HFC blend, R-404A, during in-tube evaporation in both a smooth tube and a micro-fin tube. These data presentations and comparisons are made over a mass flux range of 125 to 375 kg/m^{2}s at 0, 1 and 2% lubricant concentrations (however, alkylbenzene data were not taken at 1% concentrations). These measurements were performed at 0°C.

Effect of Lubricant Types

Figures 2 and 3 presents the results of all three lubricants at 2% lubricant concentration for the smooth tube and micro-fin tube, respectively. Also shown in both figures for comparison is the HFC-blend without lubricant (i.e., pure R-404A). Shown in both figures is a trend that is similar regardless of the lubricant-refrigerant combinations. This trend is that in-tube evaporation heat transfer coefficients increase with mass flux, with increases from about 30 to 120% going from the lowest to highest mass flux. This trend is observed regardless of the lubricant type, lubricant concentration (0, 1, or 2%), or tube type (smooth or micro-fin).

The pure refrigerant case shown in Figures 2 and 3 results in the highest heat transfer coefficients over the full mass flux range. Specifically, as 2% lubricant is added, the heat transfer performance decreases for all three lubricant types. The mineral oil and the alkylbenzene showed the most decrease in heat transfer with reference to the pure refrigerant performance and, in addition, they performed similarly. Of the three lubricant addition cases, the POE had the best heat transfer performance.

As mentioned above and as shown in Figure 2, the evaporation heat transfer performance of the POE lubricant is considerably higher than the performance of the mineral oil and alkylbenzene at 2% concentration, even though it is still less than the pure refrigerant performance. Specifically, the POE in-tube heat transfer coefficients are from 100% to 30% higher than the other two lubricants at the low and high mass fluxes, respectively, for the smooth tube. For the micro-fin tube shown in Figure 3, the POE results in heat transfer coefficients 100% to 50% higher than the other two lubricants (i.e., mineral oil and alkylbenzene) at low and high mass fluxes, respectively.

Relative to the pure refrigerant case shown in Figure 2, the performance for the POE at 2% concentration is about 70% of the pure refrigerant performance over the whole mass flux range for the smooth tube compared to a mineral oil and alkylbenzene performance that is 25 to 50% of the pure refrigerant performance. For the micro-fin tube, the performance of the 2% POE as shown in Figure 3 is about 50 and 75% of the pure refrigerant performance at low and high mass fluxes, respectively. The performance of the mineral oil and alkylbenzene is considerably less, being about 25 and 50% of the pure refrigerant performance for the low and high mass fluxes, respectively.

A comparison of the effects of lubricant types were also made at the 1% lubricant concentration, similar to the comparison at 2% lubricant concentration. In-tube evaporation heat transfer data at the 1% lubricant concentration are shown in Figures 4 and 5 for the smooth tube and micro-fin tube, respectively. Data were not taken for the alkylbenzene at 1% concentration and, therefore, these results are not shown in Figure 4 for the smooth tube and Figure 5 for the micro-fin tube. Figures 4 and 5 show trends similar to Figures 2 and 3. Specifically, as 1% lubricant is added to the refrigerant, then the evaporation heat transfer performance decreases in both the smooth tube and the micro-fin tube. However, as previously shown, the decrease in performance is much less for the 1% POE relative to the 1% mineral oil.

The 1% mineral oil performance for the smooth tube shown in Figure 4 is about 50% of the pure refrigerant performance while the 1% POE is about 80% of the pure refrigerant performance. Relative to each other, the POE lubricant addition results in heat transfer coefficients anywhere from 150% to 50% higher than the mineral oil addition results, depending on the mass flux.

For the micro-fin tube with 1% lubricant shown in Figure 5, the POE is about 80% of the pure refrigerant performance over all mass fluxes while the mineral oil is only about 40 to 50% of the pure refrigerant performance. Relative to each other, the performance of the POE refrigerant mixture for the micro-fin tube is about 50% higher than the performance of the mineral oil refrigerant mixture over a wide range of mass fluxes.

Effect of 0, 1, and 2% Lubricant Concentrations

The effects of lubricant concentrations of 0, 1, and 2% on evaporation heat transfer can be determined for both the POE lubricant and the mineral oil. For the POE lubricant in the smooth tube, data from Figures 2 and 4 shows that the that the evaporation heat transfer coefficients decrease as the concentration of lubricant increases. Specifically, for the smooth tube, pure refrigerant heat transfer coefficients decrease by 20% when 1% POE is added and by 30% when 2% POE is added. For the micro-fin tube with POE shown in Figures 3 and 5, pure refrigerant heat transfer coefficients decrease by 20% when 1% POE is added and by 25 to 50%, depending on the mass flux, when 2% POE is added.

For the mineral oil in the smooth tube, Figures 2 and 4 show that evaporation heat transfer coefficients decrease by 55% when 1% mineral oil is added and by 60%, depending on the mass flux, when 2% mineral oil is added. For the micro-fin tube in Figures 3 and 5, the mineral oil results in decreases of 40 to 60% and 50 to 75%, depending on the mass flux, for the 1% and 2% oil concentrations, respectively. As can be seen from the above results for the mineral oil, there is a large decrease in the heat transfer as 1% mineral oil is added, and then the effects of doubling the oil concentration to 2% results in only minor additional decreases in heat transfer coefficients.

CONCLUSIONS

For the HFC blend R-404A, a POE lubricant exhibited better in-tube evaporation performance than either a mineral oil or an alkylbenzene when added to the refrigerant at 1% and 2% concentrations. In addition, the higher the oil concentration, then the greater is the effect on heat transfer coefficient. The micro-fin tube exhibited greater differences from the oil effect than the smooth tube.

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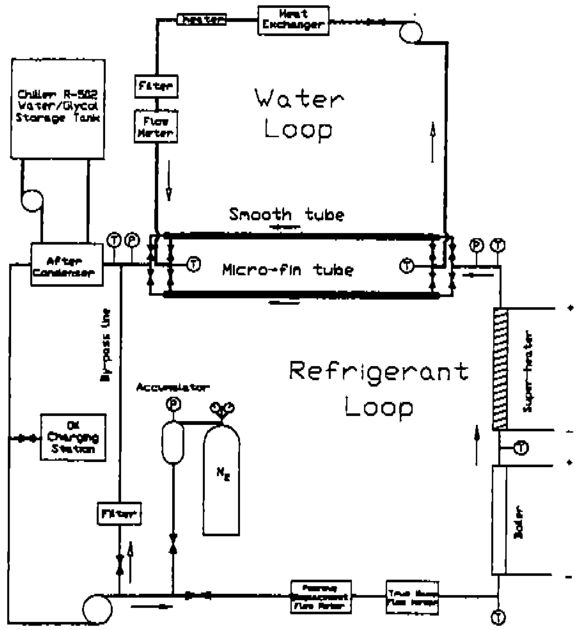

Figure 1 Schematic of in-tube evaporation test facility

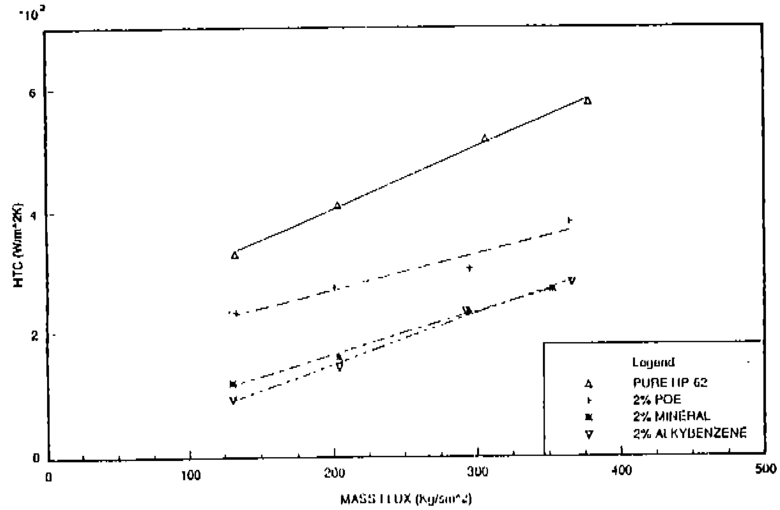

Figure 2 Comparison of smooth tube evaporation heat transfer coefficients for HFC-blend R-404A, both pure and mixed with 2% lubricant

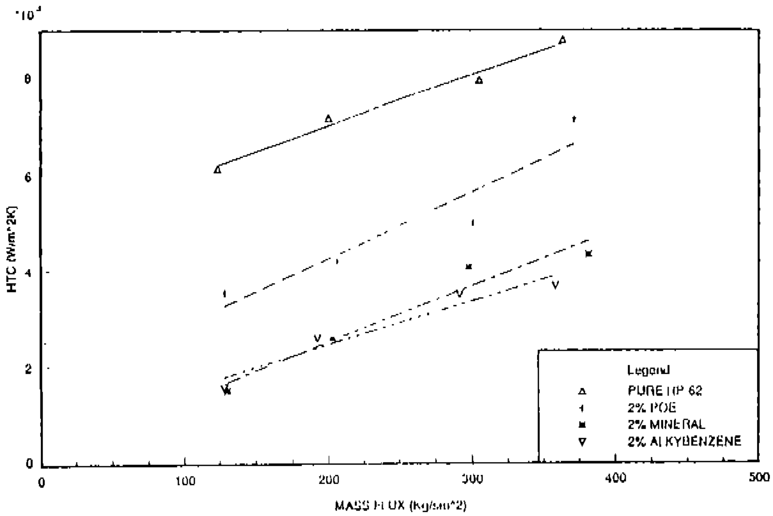

Figure 3 Comparison of micro-fin tube evaporation heat transfer coefficients for HFC-blend R-404A, both pure and mixed with 2% lubricant

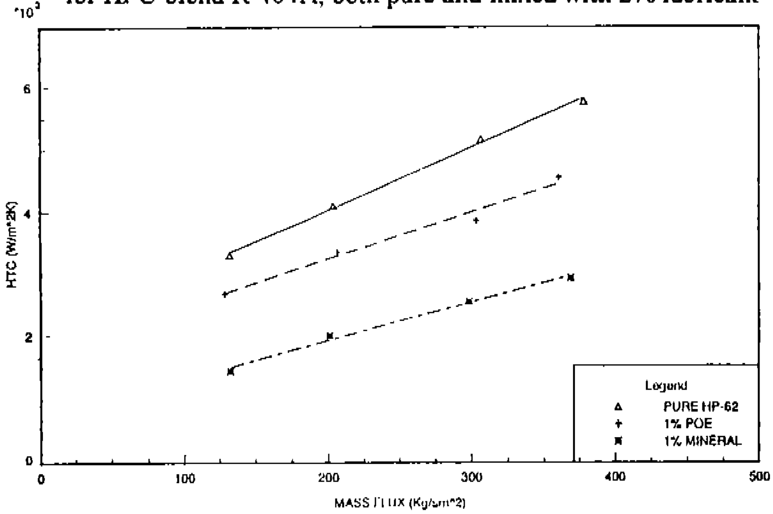

Figure 4 Comparison of smooth tube evaporation heat transfer coefficients for HFC-blend R-404A, both pure and mixed with 1% lubricant

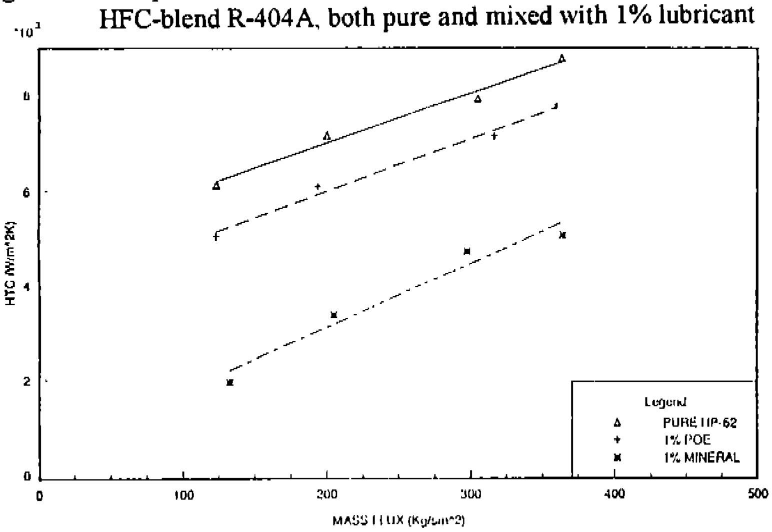

Figure 5 Comparison of micro-fin tube evaporation heat transfer coefficients for HFC-blend R-404A, both pure and mixed with 1% lubricant