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# A COMPARISON OF THE EFFECTS OF POE AND MINERAL OIL LUBRICANTS ON THE IN-TUBE EVAPORATION OF R-22, R-407C AND R-410A

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## ABSTRACT

Chlorine-free HFC refrigerants, such as R-407C and R-410A, are being considered as viable alternatives for R-22 refrigerant. Heat transfer coefficients are important considerations to meet the performance needs of the heat exchanger. Heat transfer is affected by the type and amount of lubricant circulating. This paper discusses the results of in-tube evaporation tests of R-22 with mineral oil and polyolester lubricants and compares them with R-407C/mineral oil, R-407C/POE, R-410A/mineral oil and R-410A/POE. The results show that with HFC refrigerants, namely R-407C and R-410A, the polyolester lubricants exhibit superior heat transfer characteristics in relation to mineral oils.

## INTRODUCTION

HFC refrigerant mixtures are being evaluated as replacements for R-22. In this study R-22, R-410A {R-32(50%)/R-125(50%)}, and R-407C {R-32(23%)/R-125(25%)/R-134a(52%)} were tested for in-tube heat transfer coefficients during evaporation at 7 °C. All three refrigerants were tested with lubricant concentrations of 0.2 and 2% with both POE and mineral lubricants. The refrigerants were tested in both a 3/8" O. D. smooth tube and a 3/8" O. D. micro-finned tube. The effects of the lubricant on the refrigerants heat transfer coefficients are compared for each refrigerant.

## TEST FACILITY

The test facility consists of four main parts: a refrigerant loop, a water loop, a data acquisition system, and a dual test section. For these tests only one side of the dual test section was used. The following sections provide 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 section consists of a horizontal test tube and a surrounding annulus. The inner tube is a 3/8" O.D. copper tube which is 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 tube is centered in the annulus by a series of spacers. The spacers are constructed of three stainless steel rods spaced 120 degrees about the annulus. The spacers are held in place by a series of industrial PG Teflon glands.

The test section is instrumented with temperature and pressure sensors. The temperatures are measured with resistance type temperature probes, RTDs, which have been calibrated to an accuracy of  $\pm 0.05^\circ\text{C}$ . The pressure is measured with a calibrated strain-gage type pressure transducer which is accurate to  $\pm 9$  kPa. The pressure drop in the test section is measured with a strain-gage type differential pressure transducer accurate to  $\pm 0.2$  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 sub cools the refrigerant leaving the test section. The water-glycol mixture for the heat exchanger is provided by an R-502 chilling unit. After being sub cooled, the refrigerant is circulated by a positive displacement pump. The pressure in the test section is controlled by a 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 current. The heater is electrically isolated from the rest of the system by a high pressure rubber hose. The refrigerant mass flow rate is measured by a coriolis type mass flow meter accurate to 0.4%.

### 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 an electric heater. The water mass flow rate is also measured by a coriolis type flow meter with an accuracy of 0.3%.

### 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 an 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% and 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.

$$Q_w = M_w \cdot C_{p_w} \cdot (T_{w,out} - T_{w,in}) \quad (1)$$

The quality change in the test section is determined from the energy change of the water side.

$$\Delta X = Q_w / (M_r \cdot h_{fg}) \quad (2)$$

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 resulting from the Wilson plot technique is a Dittus-Boelter type equation. The overall heat transfer coefficient is determined from the energy balance on the test section.

$$U_o = Q_w / (A_o \cdot \text{LMTD}) \quad (3)$$

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

$$\text{LMTD} = (\Delta T_1 - \Delta T_2) / \ln (\Delta T_1 / \Delta T_2) \quad (4)$$

where

$$\Delta T_1 = T_{r,out} - T_{w,in} \quad (5)$$

$$\Delta T_2 = T_{r,in} - T_{w,out} \quad (6)$$

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

$$h_i = 1 / (1/U_o - 1/h_o) A_i/A_o \quad (7)$$

## DISCUSSION OF RESULTS

Heat transfer coefficients were measured for R-22, R-410A and R-407C during evaporation at 7 °C. Each refrigerant was tested in both a smooth tube and a micro-fin tube at four mass fluxes, namely, 125, 200, 300 and 375 kg/s-m<sup>2</sup>.

### Results for Pure Refrigerants

The results for these refrigerants are plotted in Figures 2 through 7 with the pure refrigerant and both lubricant types and concentrations on each graph. From the graphs certain general trends can be seen. For example, the heat transfer coefficients increase with increasing mass flux and the heat transfer coefficients for the smooth tube is less than the micro-fin tube.

The heat transfer coefficients for R-410A are the highest of the three refrigerants tested. For the pure refrigerant cases, the smooth tube with R-410A exhibited heat transfer coefficients 25% larger than R-22 and 72% larger than R-407C. For the micro-fin tube R-410A had heat transfer coefficients 29% larger than R-22 and 100% larger than R-407C.

### Results for Refrigerants/lubricant Mixtures

The refrigerants were tested with 0.2% and 2.0% lubricant concentrations of POE and mineral oil. The results for the refrigerant/lubricant mixtures are plotted in the same figures as the pure refrigerant results, namely Figures 2 through 7.

#### **R-22**

For both the smooth tube and the micro-fin tube with R-22, the heat transfer coefficients for the mineral oil case were superior to the POE case. Compared to the pure refrigerant, the addition of either POE lubricant or mineral oil with R-22 resulted in a decrease in the heat transfer coefficient.

In the smooth tube at 0.2% lubricant concentration the mineral oil showed a 12% decrease in heat transfer coefficients while the POE showed a 25% decrease in heat transfer coefficients. For the 2.0% lubricant concentration the mineral oil case exhibited a 15% decrease in heat transfer coefficients while the 2.0% POE case showed a decrease of 31%.

In the micro-fin tube at a 0.2% lubricant concentration the mineral oil showed a 8% decrease in heat transfer coefficients while the 0.2% POE case had a decrease of 17%. For the 2.0% lubricant concentration the mineral oil case exhibited a 5% decrease in heat transfer coefficients while the 2.0% POE case had a decrease of 16%.

#### **R-407C**

For both the smooth tube and the micro-fin tube with R-407C, the heat transfer coefficients for the POE lubricant case were superior to the mineral oil case. In general, Compared to the pure refrigerant, the addition of either POE lubricant or mineral oil with R-407C resulted in a decrease in the heat transfer coefficient. The exception is the 0.2% POE lubricant concentration in the finned tube where there was a 2% increase in the heat transfer coefficient.

In the smooth tube at 0.2% lubricant concentration the mineral oil showed a 8% decrease in heat transfer coefficients while the POE showed a 6% decrease in heat transfer coefficients. For the 2.0% lubricant concentration the mineral oil case exhibited a 27% decrease in heat transfer coefficients while the 2.0% POE case showed a decrease of 14%.

In the micro-fin tube at a 0.2% lubricant concentration the mineral oil showed a 9% decrease in heat transfer coefficients while the 0.2% POE case had an increase of 2%. For the 2.0% lubricant concentration the mineral oil case exhibited a 17% decrease in heat transfer coefficients while the 2.0% POE case had a decrease of 11%.

#### **R-410A**

For both the smooth tube and the micro-fin tube with R-410A, the heat transfer coefficients for the POE lubricant case were superior to the mineral oil case. Compared to the pure refrigerant, the addition of either POE lubricant or mineral oil with R-410A resulted in a decrease in the heat transfer coefficient.

In the smooth tube at 0.2% lubricant concentration the mineral oil showed a 32% decrease in heat transfer coefficients while the POE showed a 17% decrease in heat transfer coefficients. For the 2.0% lubricant concentration the mineral oil case exhibited a 52% decrease in heat transfer coefficients while the 2.0% POE case showed a decrease of 44%.

In the micro-fin tube at a 0.2% lubricant concentration the mineral oil showed a 31% decrease in heat transfer coefficients while the 0.2% POE case had a decrease of 16%. For the 2.0% lubricant concentration the mineral oil case exhibited a 54% decrease in heat transfer coefficients while the 2.0% POE case had a decrease of 33%.

### CONCLUSIONS

Heat transfer coefficients were approximately 86% larger in the micro-fin tubes than in the smooth tubes. R-410A showed higher heat transfer coefficients than R-407C for both smooth and micro-finned tubes. R-410A is more significantly effected by the lubricants than R-407C. The maximum degradation in heat transfer coefficients due to lubricant effect for R-410A was 54% while it was 27% for R-407C. Since R-410A is more significantly effected by the lubricants than R-407C and R-410A pure has a much larger heat transfer coefficients than R-407C the nearly two to one ratio in the heat transfer coefficients for the pure refrigerants is reduced for the lubricant cases.

In summary, the results show that with HFC refrigerants, namely R-407C and R-410A, the polyolester lubricants exhibit superior heat transfer characteristics in relation to mineral oils.

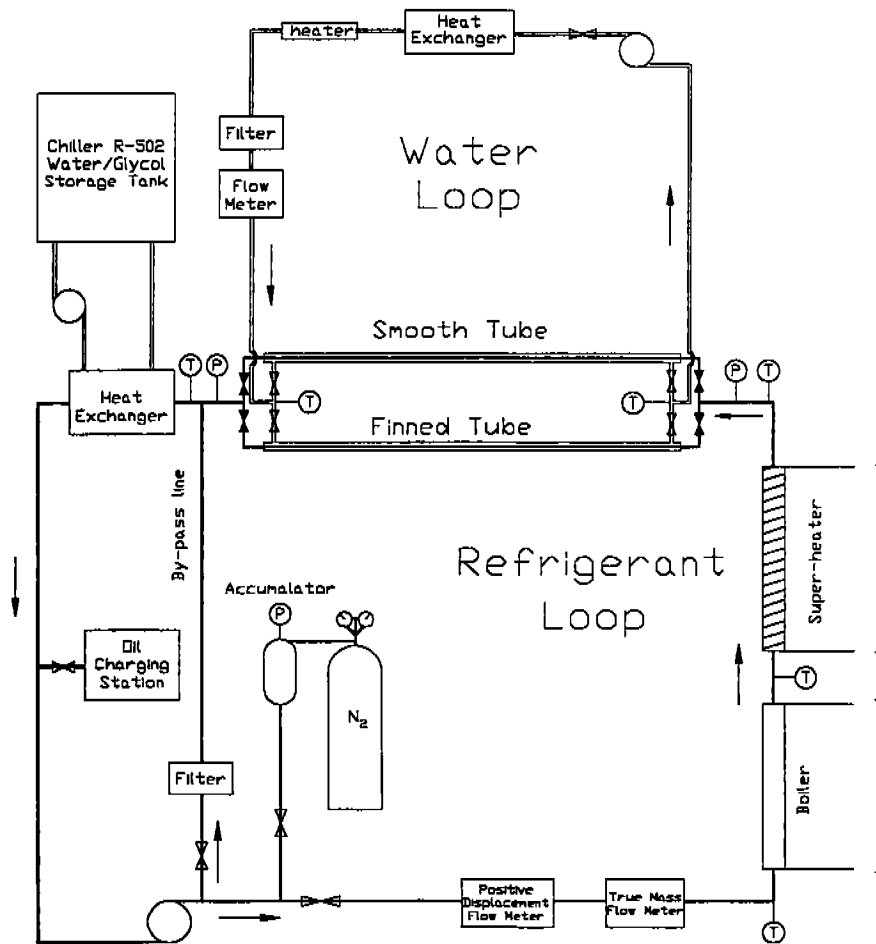
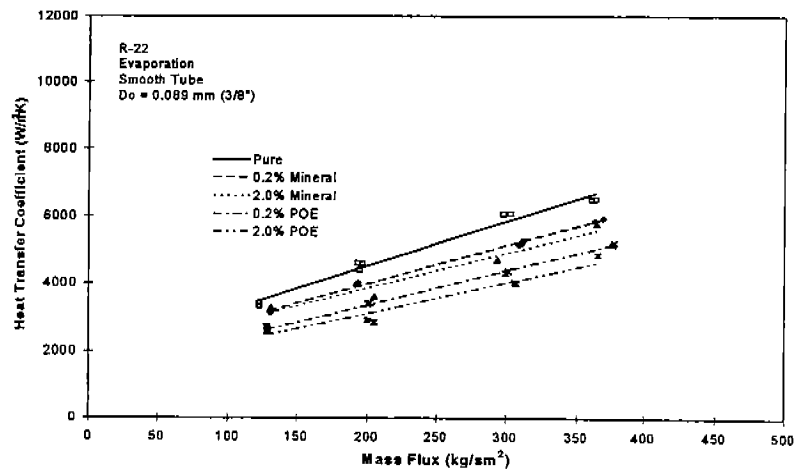
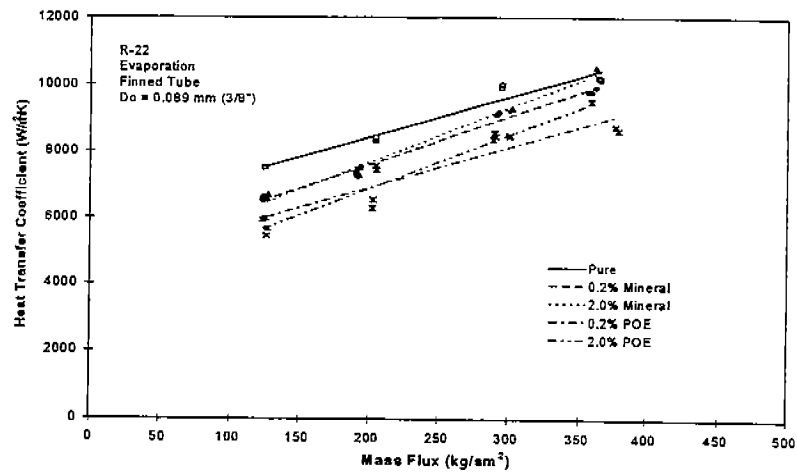


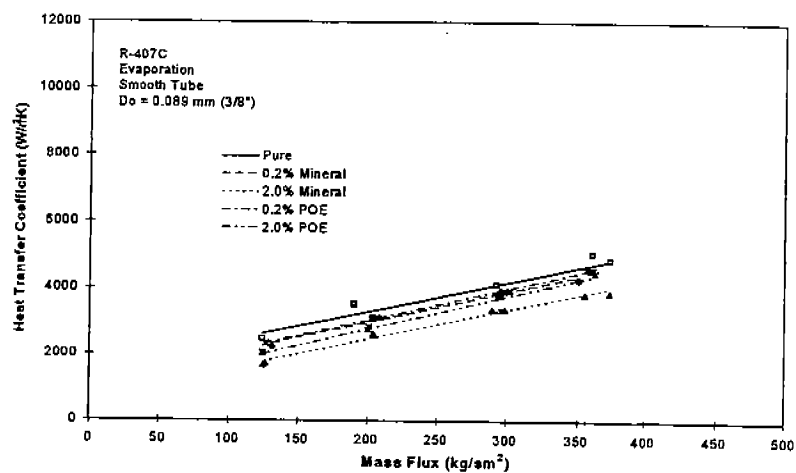
Figure 1 Test Facility Schematic



**Figure 2** Comparison of smooth tube evaporation heat transfer coefficients for R-22 both pure and mixed with lubricant



**Figure 3** Comparison of micro-fin tube evaporation heat transfer coefficients for R-22 both pure and mixed with lubricant



**Figure 4** Comparison of smooth tube evaporation heat transfer coefficients for R-407C both pure and mixed with lubricant

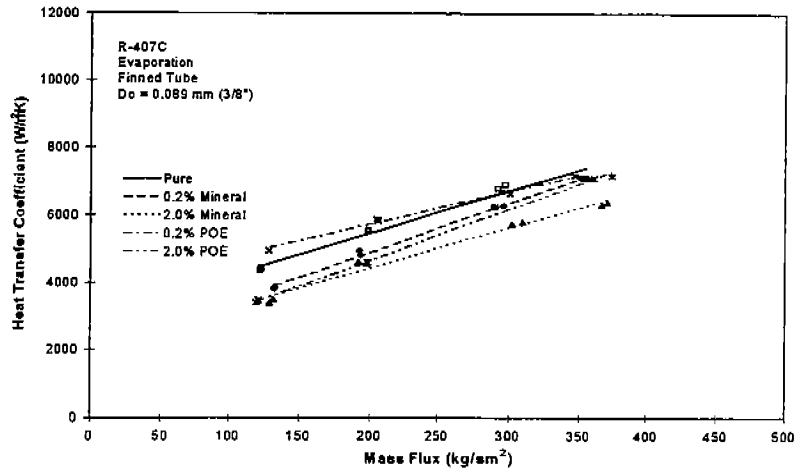


Figure 5 Comparison of micro-fin tube evaporation heat transfer coefficients for R-407C both pure and mixed with lubricant

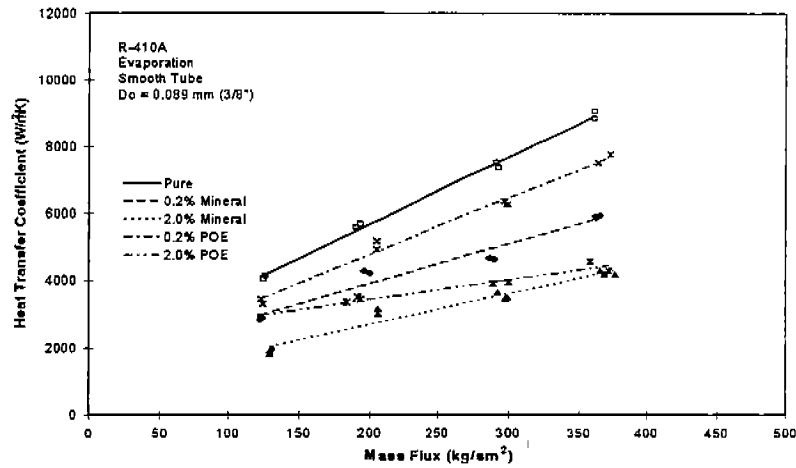


Figure 6 Comparison of smooth tube evaporation heat transfer coefficients for R-410A both pure and mixed with lubricant

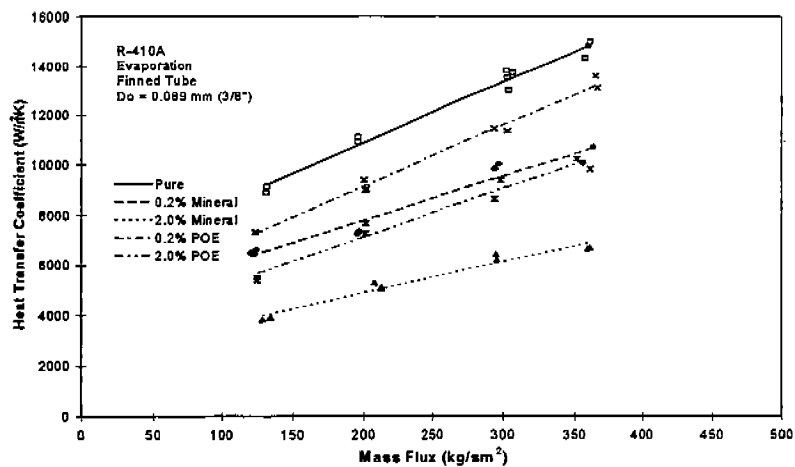


Figure 7 Comparison of micro-fin tube evaporation heat transfer coefficients for R-410A both pure and mixed with lubricant