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FLOW CONDENSATION OF PURE AND OIL CONTAMINATED REFRIGERANT HFC134a IN A HORIZONTAL TUBE

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The paper reports the results of condensation heat transfer and pressure drop from tests with pure and oil contaminated refrigerant HFC134a in a horizontal tube (10 m in length, 6 mm ID). The heat transfer coefficient in case of oil contaminated refrigerant is shown to depend strongly on the definition of the saturation temperature. Using the *pure refrigerant saturation temperature* (hence disregarding the influence of oil on the vapor pressure), the results in average heat transfer coefficient show only minor effect of the oil contents. If the *saturation temperature of the refrigerant+oil mixture* is used, there is thus a considerable degradation of the heat transfer coefficient (as expected) with increasing oil concentrations.

INTRODUCTION

In a vapor compression refrigeration and heat pump system, a common problem is that small amounts of compressor lubricant, as a part of the working fluid, is circulated with the refrigerant. This migrated oil affects the performance of the condenser and evaporator.

The influence of oil on the condensation has been shown in a number of publications. **Tichy** et al (1985) reported 10% to 23% reduction in CFC12 heat transfer coefficients, and 2% to 6% increase in pressure drop, at 2% and 5% 300 SUS naphthenic base oil concentrations. **Schlager** et al (1988) examined the condensation of HCFC22 and a 150 SUS naphthenic oil mixture, and found a 13% decrease in heat transfer with a 5% lubricant content, and that the pressure drop is decreased or remained approximately constant despite of the addition of lubricant. **Sur** and **Azer** (1991) demonstrated that the presence of 1.2%, 2.8% and 4.0% oil (150 SUS naphthenic oil) reduced the heat transfer coefficients by 7%, 12%, and 16%, respectively, compared with pure refrigerant CFC113, but no visible effect of oil on pressure drop. **Eckels** and **Pate** (1991) tested refrigerant+oil mixtures of CFC12 and HFC134a. They showed a rough 10% degradation in heat transfer for a CFC12 and 5% 150 SUS naphthenic oil mixture, but no significant effect of 165 SUS PAG oil on heat transfer coefficient of HFC134a. They also pointed out that the presence of 165 SUS PAG oil resulted in pressure drop increasing for HFC134a by approximately 20% to 40%, but no considerable influence registered for CFC12 with 150 SUS naphthetic oil. It needs to be emphasized that different definitions of saturation temperature of the mixture were used by different investigators to calculate heat transfer coefficient. This affects the results of the heat transfer coefficient of the mixture. Some discussions on this point will be given later.

The paper here is to enhance the understanding of condensation for refrigerant+oil mixtures inside tubes. The results of heat transfer and pressure drop from the testing with HFC134a and an ester-based oil are presented, and compared with data from tests with oil-free HFC134a.

EXPERIMENTAL FACILITY

The testing apparatus used is almost the same as the one in previous work (**Shao** 1993b,c) except some necessary revisions, which enable a study of lubricant+refrigerant mixtures, as shown in **Figure 1.** The experimental condenser consists of ten smooth concentric copper tubes linked together in series, inner tube 6 mm ID, 8 mm OD, outer tube 12.6 mm ID, each 1m of length. The refrigerant flows through the inside tube, and the cooling water through the annulus in the opposite direction.

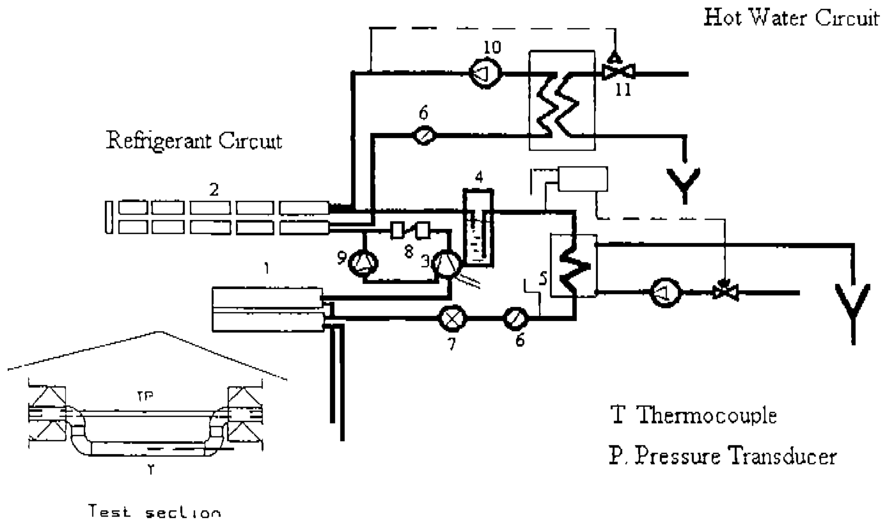


Figure 1 *A schematic drawing of the experimental facility: 1 Evaporator, 2 Condenser, 3 Compressor, 4 Receiver, 5 Subcooler, 6 Flow meter, 7 Expansion valve, 8 Oil separators, 9 Oil pump, 10 Water pump, 11 Thermostatic water valve*

Along the condenser, eleven "test sections" are positioned at the inlet and the exit of the test tube and between the sub-sections, where the temperatures of refrigerant and cooling water and local pressure can be measured. The refrigerant temperatures are measured by four thermocouples (Type T) positioned around the circumference of the tube surface under the adiabatic condition. At the same time, four thermocouples (Type T) are inserted into the condenser at "test sections" 4, 7, 9, and 11 to measure the refrigerant temperature in a different way. This gives a possibility to compare the refrigerant temperatures mentioned from two different temperature measurements. The cooling water temperature is obtained by a thermocouple (Type T) inserted into the bend tube (corresponding to a mixing chamber, which enable an average water temperature to be determined, see **Figure 1**). The measured temperatures and pressures are transferred to a computer for data treatment through a data logger. The average temperatures and pressures during a steady period of time are used to calculate heat transfer coefficients.

Glass tubes (10 cm in length) are placed at the "test sections" 2, 5, 8 and 10, so that visual observations can give an overview of flow patterns change along the condenser. The inside diameter of the glass tube is the same as that of the test copper tube, so as not to disturb condensate flow. Two oil separators are employed to ensure oil-free conditions when testing pure refrigerant. When testing oil+refrigerant mixtures, the oil in a finely atomized form is injected before the condenser by aid of an oil pump, as shown in **Figure 1.** The amount of oil injected is determined by removing a sample of the mixture after the condenser and boiling off the refrigerant. The refrigerant mass flow rate is measured by a coriolis mass flow meter with an accuracy of ± 0.2%. The flow of the cooling water is measured by a volume flow meter of turbine type, accuracy of ± 2%. The accuracies of measured pressure and absolute temperature are 0.1% and about ± 0.2°C, respectively. The error of the temperature difference is estimated to be approximately ± 0.05°C.

DATA REDUCTION

The heat transfer coefficients are expressed in form of local α as well as mean $\bar{\alpha}$ values. The local one is defined as:

$$\alpha = \frac{q}{t_s - t_w} \quad (1)$$

where, q denotes heat flux (W/m^2), t_s denotes saturation temperature (°C), and t_w denotes wall temperature (°C). The inner wall temperature t_w is calculated from the heat flux, temperature of cooling water and so on. The heat transfer coefficient on the cooling water side was showed to be determined with good accuracy using the well-known Dittus-Boelter-type equation (**Shao** 1993b).

The local heat flux q at the "test section" is calculated by taking the mean value of the average heat flux in the sub-section before and after the "test section". The average heat flux in the sub-section can be written as:

$$\bar{q} = \frac{Q_j}{\pi \cdot d_i \cdot \Delta l} = \frac{(c_{pcw} \cdot m_{cw} \cdot \Delta t_{cw})_j}{\pi \cdot d_i \cdot \Delta l} \tag{2}$$

where, Δl = length of each sub-section = 1 m, d_i = inner diameter of the tube (m)

Q_j = heat transferred in No.j sub-section (j=1,..,10)

c_{pcw} = specific heat of the water (J/kg K), m_{cw} = mass flow rate of cooling water (kg/s)

Δt_{cw} = temperature change of the cooling water in a sub-section (K)

The mean heat transfer coefficient is defined as:
$$\bar{\alpha} = \frac{Q}{\pi \cdot d_i \cdot L_e \cdot (\bar{t}_s - \bar{t}_w)} \tag{3}$$

where, $Q = c_{pcw} \cdot m_{cw} \cdot \Delta t_{cw}$ = heat transferred in the L_e length, the effective heat transfer length L_e means that only two-phase condensing regions are taken into consideration when calculating condensation mean heat transfer coefficients. For simplicity, L_e = 7m is here taken, this corresponds to vapor quality of approximately inlet: 0.88-0.95; outlet: 0.1-0.2. $\bar{t}_s$ and $\bar{t}_w$ are arithmetic average of t_s and t_w within the L_e length.

The saturation temperature t_s of pure HFC134a is determined by the local pressure. For the mixtures of refrigerant+oil, the influence of oil on the saturation temperature was usually neglected (**Eckels** et al 1991, **Schlager** et al 1988). In the work by **Tichy** et al (1985), they used a measured value from the thermocouple inside the condenser as saturation temperature, and in an investigation by **Sur** and **Azer** (1991), where they applied Raoult's law to calculate the saturated vapor pressure of the lubricant+refrigerant mixture. The author made a comparison of results from the measurement (**Shao** et al 1992, 1993a) and the Raoult's law, and found a deviation up to 25% in vapor pressure (**Shao** 1993d).

The saturation temperature of oil+refrigerant mixture is higher than that of pure refrigerant at the same pressure. The temperature difference is defined as "superheat temperature of oil+refrigerant mixture". In a condenser or evaporator, this "superheat temperature" depends on vapor quality and bulk oil content in a system at a certain pressure. **Figure 2** gives an example of HFC134a and an ester-based oil from the measurement (**Shao** 1993a). On the other hand, the temperature difference between t_s and t_w is mostly 2-4 K. This "superheat temperature" may therefore cause a big influence on heat transfer coefficient. In case of calculating heat transfer coefficient of oil+HFC134a mixture, the saturation temperature t_s will here be taken from both real saturated temperature of the mixture from **Shao** (1993a), and pure refrigerant relation (i.e. assuming no influence of oil on vapor pressure), respectively, the results from two such cases are labeled as α_a and α_b, and then compared with each other.

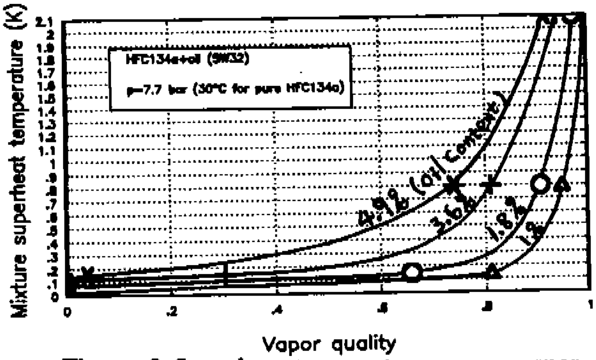


Figure 2 *Superheat temperature versus vapor quality at different bulk oil contents (mass %)*

RESULTS AND DISCUSSION

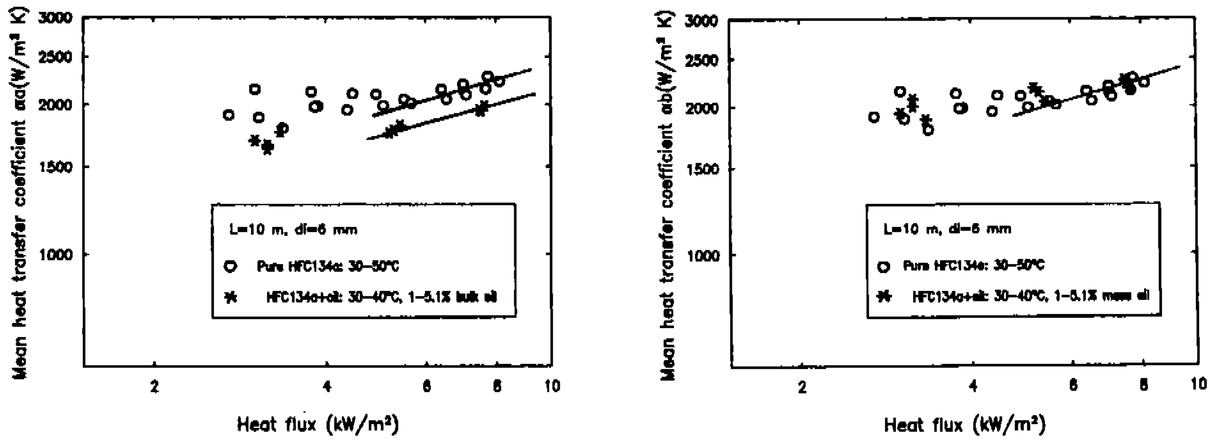
A series of tests have been conducted with both pure HFC134a and oil+HFC134a mixtures. The oil is injected to form a mixture of oil+refrigerant before the condenser when testing with HFC134a+oil mixtures. The oil used is from Castrol Ltd. U.K.. It has a viscosity of 32 cSt at 40°C, and a viscosity index of 118. The test ranges in this investigation are condensing temperatures 30-50°C, mass flux 120-290 kg/m^2s, oil contents 0-5.1% by mass.

Heat transfer coefficient and pressure drop from the tests with pure HFC134a and oil+HFC134a mixtures are presented below. The effect of oil on refrigerant enthalpy was demonstrated to be negligible in the region of interest (**Shao** 1993d). The heat balance error from the inlet to the exit of the condenser is less than ± 2%. Thermodynamic properties are taken from the work by **Morrison** et al (1991).

Heat Transfer Coefficient

Results of measurement of the heat transfer are presented as absolute values of heat transfer coefficient from the experiments with pure HFC134a and oil+HFC134a mixture, as well as ratios of heat transfer coefficient between oil+refrigerant mixture and pure refrigerant. In case of oil contaminated refrigerant, saturation temperature of the mixture is taken from the mixture measurement t_s (**Shao** 1993a) and pure refrigerant relation, respectively, to calculate heat transfer coefficients. The results α_a and α_b are given in form of the figures (a) and (b) in the following descriptions.

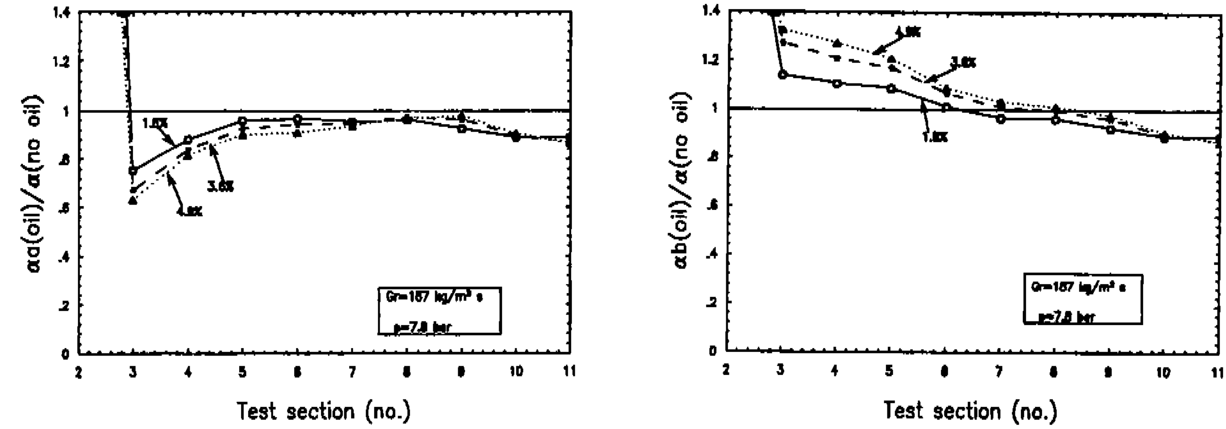
A direct comparison of mean heat transfer coefficient between HFC134a and HFC134a+oil is performed, as demonstrated in **Figure 3**, where the mean heat transfer coefficient is plotted versus heat flux. The **Figure 3** (a) indicates a degradation of 11% and 20% in oil+refrigerant heat transfer coefficient $\bar{\alpha}_a$, at 2% and 5% oil concentrations, compared with pure refrigerant, but no degradation in heat transfer coefficient $\bar{\alpha}_b$ with the addition of oil in **Figure 3** (b). It also indicates the relation between heat transfer coefficient and heat flux. The heat transfer coefficients seem independent of heat flux at less than approximately 5 kW/m^2, whereas it is a function of the heat flux above 5 kW/m^2. A visual observation indicates that the flow pattern is annular at the region larger than 5 kW/m^2, but the stratified flow prevails at the region less than 5 kW/m^2.



(a) Based on mixture t_s *(b) Based on pure refrigerant relation*

Figure 3 *Mean heat transfer coefficient versus heat flux for pure HFC134a and HFC134a+oil (SW32)*

Local heat transfer coefficients are also compared between oil+refrigerant mixture and pure refrigerant, as shown in **Figure 4**. It is seen in **Figure 4** (a),(b) that the heat transfer coefficient of oil+refrigerant is higher than that of pure refrigerant at an entrance region (test section 2). This is due to the fact that the pure refrigerant does not condense at the "test section" 2, in other words, is in superheat vapor state, whereas the annular flow is observed at "test section" 2 for oil+refrigerant mixture. **Figure 4** (a) indicates that the heat transfer coefficient α_a is reduced considerably, especially at the beginning of the condensing section owing to the oil-rich liquid phase. **Figure 4** (b) shows an increase at first half and a decrease at the second half of the condenser in heat transfer coefficient α_b of the mixture, compared with pure refrigerant. This explains why the mean heat transfer coefficients do not change more as shown in **Figure 3** (b).



(a) Based on mixture t_s *(b) Based on pure refrigerant relation*

Figure 4 *Relative local heat transfer coefficient between HFC134a+oil and pure HFC134a at different bulk oil mass concentrations*

Pressure Drop

As stated before, the effect of oil on the pressure drop of refrigerant is reported differently in several investigations. Some results from the present work are shown in **Figure 5**, in form of the ratio between the pressure drop for oil+refrigerant mixture and pure refrigerant as function of oil concentrations. The enhancement of pressure drop of oil+refrigerant mixture is on the average about 20% within the tested oil percentages, compared with pure refrigerant.

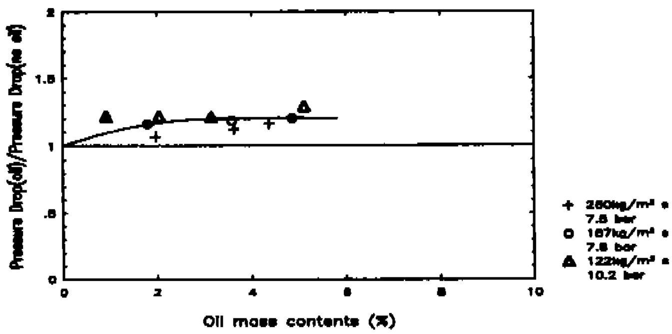


Figure 5 *Relative pressure drop between HFC134a+oil mixture and pure HFC134a versus bulk oil mass contents*

CONCLUSIONS

The tests with pure HFC134a and oil contaminated HFC134a have been carried out. Data of heat transfer and pressure drop are presented.

The conclusion of the heat transfer coefficient in case of oil contaminated refrigerant is shown to depend strongly on the definition of the saturation temperature. This should be paid more attention when comparing results from different investigations in the literature. The conclusions are here given according to two different definitions of the saturation temperature. From the practical view of point, the definition based on pure refrigerant relation (here referred to as α_b) seems more straightforward and simple to use.

In case of using the saturated temperature for pure refrigerant, the heat transfer coefficients α_b of the mixture (refer to figures (b)) are not found to be reduced noticeably, compared to results with pure refrigerant. This result is explained in the paper.

If, however, the saturation temperature for the mixture is used, the condensation heat transfer coefficients α_a (refer to figures (a)) of oil+refrigerant mixture are found to be reduced by on the average 11%-20%, depending on oil concentrations, compared to results for pure refrigerant. The degradation of heat transfer coefficient, which can be expected, is explained by the increased liquid viscosity of oil+refrigerant mixture, which decreases the molecular and turbulent transport in the condensate film.

The pressure drop of oil+HFC134a is increased by 20% in this investigation, in comparison to pure HFC134a. The influence of oil on pressure drop may strongly depend on the properties of oil and refrigerant, which can explain different conclusions in the open literature.

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