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An Experimental Comparison of Evaporation and Condensation Heat Transfer Coefficients for HFC-134a and CFC-12

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HEAT TRANSFER COEFFICIENT, TWO-PHASE FLOW BOILING OF HFC134a

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ABSTRACT

Evaporation of HFC134a inside smooth, horizontal tubes is studied. Tests are made with pure refrigerant and with oil-refrigerant mixtures. Heat flux has varied from 2 kW/m^2 to 10 kW/m^2. The inner diameter of the tubes are 12 mm. Two evaporators are used, 4 and 10 m long. Oil content is varied from 0 to 2.5 mass percentage (synthetic oil, EXP-0275). Oil free HFC134a is found to have higher heat transfer coefficient than HCFC22 at the same heat flux, as well as mass flux. The effect of oil in the refrigerant depends on the heat flux. At 2 and 4 kW/m^2, the heat transfer coefficient has a maximum value for an oil content of around 0.5 mass percentage. No increase at all is registered for a heat flux of 6 kW/m^2.

Heat transfer coefficients for pure refrigerant are also compared to existing correlations. Pierre's correlation predict values with a reasonable accuracy. The by Jung modified Chen-relation, however, overestimate the heat transfer coefficient. Discrepancies are probably mainly due to errors in thermodynamic properties.

INTRODUCTION

When it became known that chlorine in refrigerants participate in the depletion of the ozone layer, a search for alternatives started. One refrigerant which has a high ozone depletion factor is CFC12. A possible substitute for this media is HFC134a. It is necessary that as much as possible is known of the behavior of HFC134a when introducing it to the market One particular area of interest is the performance in the evaporator. This paper presents preliminary results from heat transfer coefficient measurements for two phase flow boiling in horizontal tubes. The tests are carried out for pure refrigerant as well as for a mixture of refrigerant and oil.

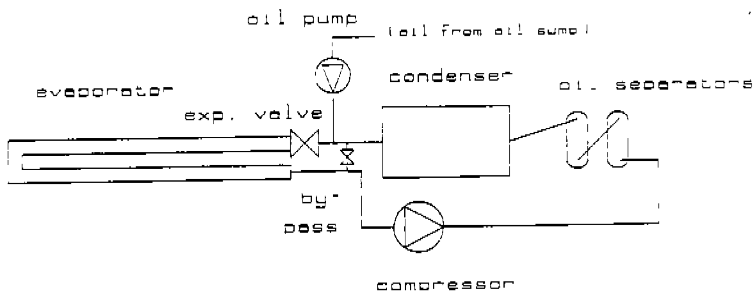
Applied power varies from 2 kW/m^2 to 10 kW/m^2. Inlet quality is around 0.25 in the long evaporator, and 0.30 in the short. Evaporation temperature has been varied from -20°C to 5°C in the oil free tests In tests with oil-refrigerant mixture, the evaporation pressure has been kept as constant as possible and close to 2.3 bar, which corresponds to an evaporation temperature of -6°C for pure refrigerant.

EXPERIMENTAL APPARATUS

The tested evaporator consists of ten smooth copper tubes linked together (inner diameter: 12 mm, length: 1 m). There are four sight glasses (same inner diameter and 10 cm long) which enable a visual study of the flow at given sections of the evaporator. The tubes are connected to each other and form a horizontal "U". The sight glasses are placed after tube 1, 4, 6 and 9. A series of tests are carried out with a shorter evaporator, 4 m.

The evaporator tubes are heated electrically and power is applied uniformly along the tubes. Mass flow of refrigerant is controlled by a manual expansion valve. It is adjusted so that all superheating takes place in the last tube only. The evaporator is connected to an ordinary open reciprocating compressor and a water cooled condenser. Two large oil separators are used to ensure an oil free evaporator. In the case of tests with oil-refrigerant mixtures, the oil is inserted before the expansion valve. The desired mass flow of oil is acquired by adjusting the length of the stroke of the oil pump.

Surface temperatures on the tubes are measured by four thermocouples placed around the circumference at the outlet of the tube. Temperature before the expansion valve and after the evaporator are measured by thermocouples inserted into the tube. Condenser pressure, evaporator pressure and pressure drop along the evaporator are also measured. Mean values of temperatures and pressures during a period of steady state are used for calculations of the heat transfer coefficient.



schematic drawing of test equipment
figure 1

HEAT TRANSFER COEFFICIENT CORRELATIONS

In this paper, the results from two phase flow boiling of HFC134a are compared to two different correlations for heat transfer coefficients. These are a correlation according to Pierre [5] and a modified Chen relation [3]. Both correlations apply to two phase flow boiling in smooth horizontal tubes. The first correlation applies to mean values of heat transfer coefficients along the evaporator, and the second to local heat transfer coefficients.

Nomenclature

- d diameter (m)
- L length (m)
- g gravitational acc. (m/s^2)
- $\dot{Q}$ heat input (W)
- $\dot{q}$ heat flux (W/m^2)
- $\dot{m}$ mass flow (kg/s)
- $\dot{G}$ mass flux (kg/(s·m^2))
- Δi enthalpy change over tested section (J/kg)
- h_{fg} latent heat of vaporization (J/kg)
- x quality
- h heat transfer coefficient (W/(m^2·K))
- Δt temp diff. tube- refr. (K)
- ρ density (kg/m^3)
- μ viscosity (Pa·s)
- λ thermal conductivity (W/(m·K))
- c_p specific heat (J/(kg·K))

subscripts

- l liquid
- v vapor
- l.o. liquid only
- loc local value

Pierre's correlation

Pierre's tests include three refrigerants, namely CFC12, HCFC22 and HCFC502. The evaporator length was 4.08 m to 14.30 m, and the inner diameter varied from 12 mm to 18 mm. Applied power varied from 1.1 kW/m^2 to 30 kW/m^2. He uses the dimensionless Nusselt number, Nu, to represent the heat transfer coefficient. The Nusselt number is shown to be proportional to Reynold's number, Re, and a boiling number, K_f, introduced by Pierre. These dimensionless numbers are defined as follows:

$$Nu = \frac{h \cdot d}{\lambda}$$

$$Re = \frac{4 \cdot \dot{m}}{\pi \cdot d \cdot \mu}$$

$$K_f = \frac{\Delta i}{g \cdot L}$$

The heat transfer coefficient, h, is the mean value of the heat transfer coefficients over the length L.

Pierre found it possible to represent all tested refrigerants with the same correlation:

$$Nu = C \cdot (Re^2 \cdot K_f)^n$$

C and n are independent of the refrigerant, but depend on whether or not the media is superheated. The constant C, however, also depend on used thermodynamic properties. The following values are given by Pierre:

$$\left.\begin{aligned} n &= 0.5 \\ C &= 1.1 \cdot 10^{-3} \end{aligned}\right\} \text{without superheat}$$

$$\left.\begin{aligned} n &= 0.4 \\ C &= 1.0 \cdot 10^{-2} \end{aligned}\right\} \text{with superheat (5 to 7 K)}$$

Unfortunately, the property data for HCFC22 used in this paper differs from data used by Pierre [5]. A correction for these differences in properties can be established in the following way:

$$C_{corr} = C \cdot \left(\frac{(\lambda/\mu)_{Pierre}}{(\lambda/\mu)_{data}} \right)$$

For following coparisions, the corrected constant is used (which is around 20% lower than Pierre's constant).

A modified Chen correlation

Chen's correlation applies to two phase flow boiling of water. He divides the two-phase flow boiling heat transfer into two different forms: nucleate boiling and convective evaporation. At low qualities, nucleate boiling and convective evaporation coexist, but as quality increases, nucleate boiling is suppressed.

The dimensionless heat transfer coefficient is represented by the ratio of local heat transfer coefficient and heat transfer coefficient for liquid only, $h_{loc}/h_{l.o.}$. It is plotted versus the inverse Martinelli parameter, $1/X_{tt}$, which is proportional to quality. It is therefore possible to view the graph as local heat transfer coefficient's dependency on quality. In the area where the two forms of boiling coexist, the heat transfer coefficient ratio is almost independent of quality, but when nucleate boiling is suppressed, the ratio increases with increasing quality. Definitions are as follows:

$$X_{tt} = \left[\frac{1-x}{x} \right]^{0.9} \cdot \left[\frac{\rho_v}{\rho_l} \right]^{0.5} \cdot \left[\frac{\mu_l}{\mu_v} \right]^{0.1}$$

$$h_{l,o} = 0.023 \cdot \frac{\lambda_l}{d} \cdot \left[\frac{G\cdot(1-x)\cdot d}{\mu_l} \right]^{0.8} \cdot \left[\frac{c_{pl}\cdot\mu_l}{\lambda_l} \right]^{0.4}$$

$$h_{loc} = \frac{\dot{q}}{\Delta t}$$

Heat transfer coefficient ratio in the region of convective evaporation can be described by a correlation. The original correlation given by Chen has been modified recently by Jung to apply to refrigerants [3] The point of transition between coexisting boiling forms and suppressed nucleate boiling depends on, for example, heat flux, mass flux, refrigerant and probably surface condition. The heat transfer coefficient's dependency on mass flux and heat flux is illustrated by introducing a boiling number, Bo.

$$Bo = \frac{\dot{q}}{\dot{G}\cdot h_{fg}}$$

This can be interpreted as the ratio between mass flux of vapor generated on the heated surface ($\dot{q}/h_{fg}$) and total mass flux parallel to the surface ($\dot{G}$). An increase in boiling number results in an increase in heat transfer coefficient ratio, and the point of transition is moved towards greater qualities.

RESULTS FROM TESTS WITHOUT OIL

Local heat transfer coefficients can be calculated with the help of measured temperatures, pressures and applied power. Evaporator pressure with correction for pressure drop are used to calculate refrigerant temperature in the tubes. The methods described above have been used to represent test results in a dimensionless form.

Mean heat transfer coefficients

To compare HFC134a with known refrigerants, the mean heat transfer coefficient is plotted versus mass flux as well as heat flux. Results from tests with HCFC22 are included as a reference. These figures indicate that HFC134a has advantageous boiling heat transfer behavior. They also indicate a sudden change in magnitude of the heat transfer coefficient. This is probably due to a change in flow pattern. A visual study of the sight glass at the end of the evaporator indicate that, at high mass flux, the liquid is spread in a thin film over the entire circumference. This flow pattern is positive for the heat transfer.

In dimensionless representation, the results are compared to Pierre's correlation. If the corrected constant is used, the correlation for non superheated evaporation, predict values fairly close to the measured ones for the short tube, figure 4. Pierre's data show scattering of the same magnitude as presented in this paper. It is noticeable that the length of the evaporator seems to have some influence on the Nusselt number. Pierre's correlation for superheated media, however, does not adequately predict the heat transfer coefficient, as shown in figure 6. This is not surprising since the correlation is only valid when the superheat is 5 to 7 K and in these tests it has varied between 5 and 18 K.

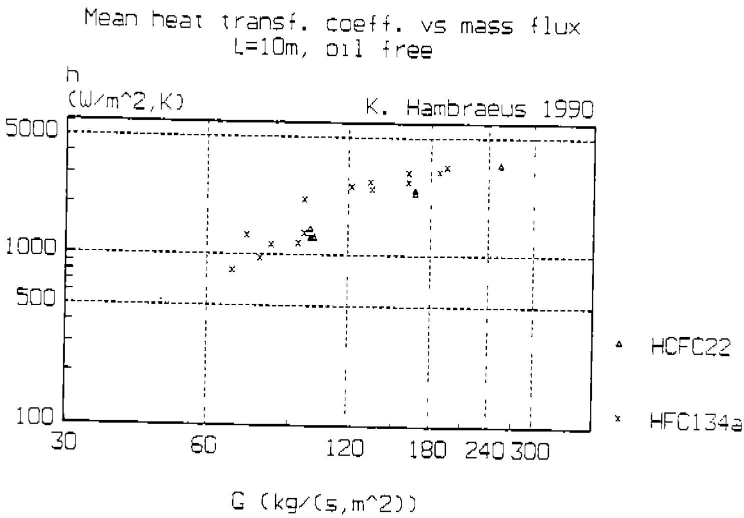


figure 3

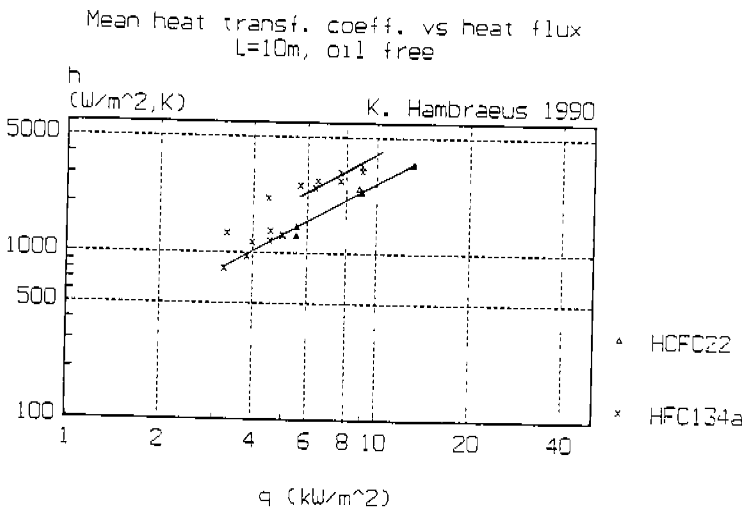


figure 4

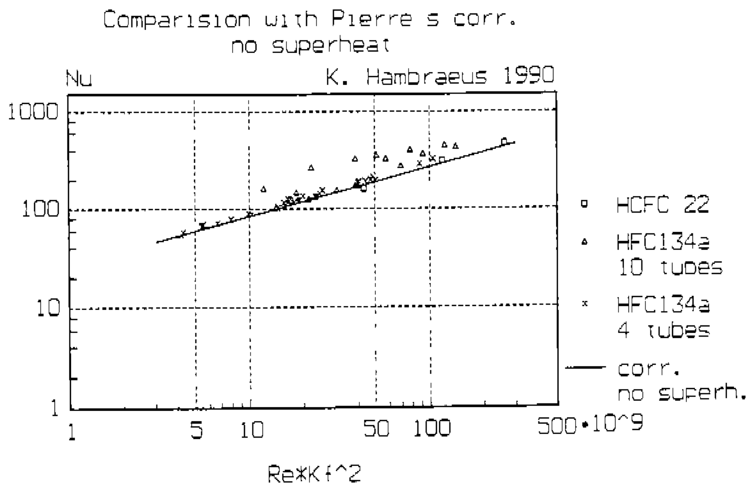


figure 5

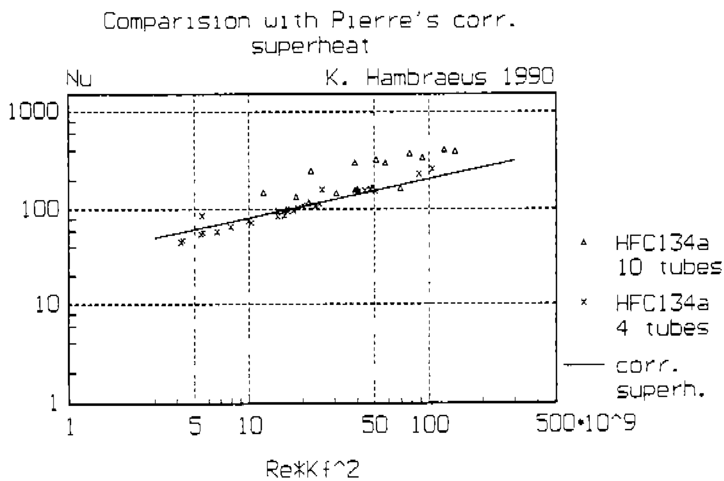


figure 6

Local heat transfer coefficients

A plot of the local heat transfer coefficient along the evaporator illustrates how it varies with quality. At low heat flux, there is almost no change of the heat transfer coefficient during the evaporation. For greater heat flux, however, there is an increase in heat transfer coefficient along the evaporation. This is due to two phase flow phenomenon.

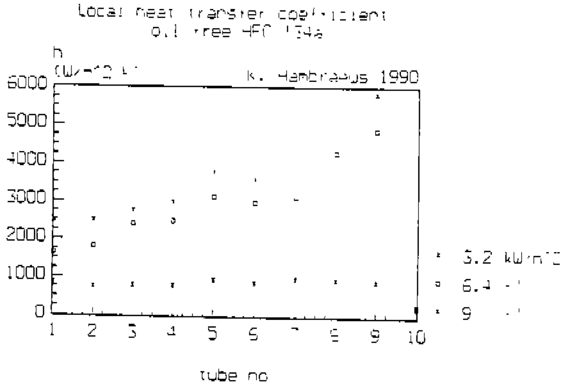


figure 7

In dimensionless representation, local heat transfer coefficients are represented by the ratio between measured heat transfer coefficient and calculated for liquid only, as described above. The correlation according to Jung does not predict heat transfer coefficients very accurately. The results do, however, follow the general trend of the correlation. The discrepancies are probably mainly due to errors in thermodynamic data.

The influence of the boiling number, Bo, is illustrated by the results from tests with evaporators of different length. Tests in the 4 m evaporator have boiling numbers which are almost three times as high as those in the 10 m long one, and the point of transition to suppressed nucleate boiling differs by a factor of two, see fig. 8.

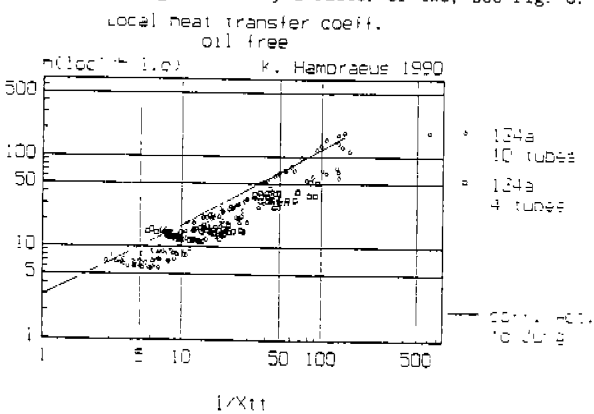


figure 8

RESULTS FROM TESTS WITH OIL

The heat transfer coefficient is measured for different amount of oil inserted in the refrigerant flow. It is compared to pure refrigerant by dividing the measured heat transfer coefficient with the heat transfer coefficient for the same heat flux, but with oil free refrigerant. The amount of oil inserted is measured by removing a sample of the mixture and boiling off the refrigerant. Both weight and volume of the sample is measured. Results are related to mass percentage oil.

The tested oil has a viscosity of 28.6 cSt at 40°C and a viscosity index of 121[1]. It is a synthetic oil and is totally soluble in HFC134a for temperatures between -50°C and 80°C. It is also fully miscible with ordinary mineral oils.

Oil influences the properties of the refrigerant markedly [1]. Since there are no data available concerning this behavior, the heat transfer coefficient is calculated with the help of measured pressure and as if it was pure refrigerant. Even though this does not produce the theoretically correct value of the heat transfer coefficient, it is the, for practical use, simplest and most straight forward procedure.

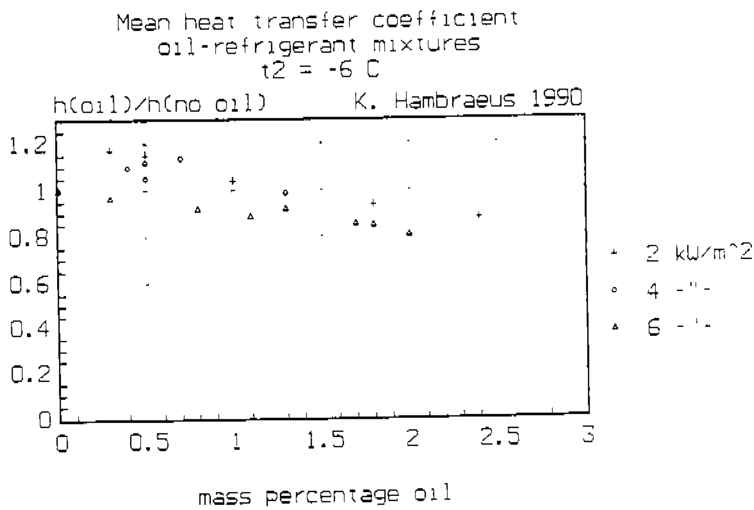


figure 9

A small amount of oil in the refrigerant seems to have a slight positive effect on the heat transfer coefficient at a heat flux lower than 4 kW/m^2. At higher heat flux, no positive effect at all is registered when introducing oil in the refrigerant. Maximum increase of heat transfer coefficient occurs at an oil content around 0.5%. At a heat flux of 2 kW/m^2, the highest registered heat transfer coefficient is around 20% better than the heat transfer coefficient for pure refrigerant (increases of 50% has been noted by other authors [7]).

[1]Oil: EXP-0275, CPI Eng. Services Inc.

For higher heat flux, there is a continous decrease in heat transfer coefficient with increasing oil content (20% lower at 2 mass percentage oil).

Local heat transfer coefficient for tests with oil is also compared to the local heat transfer coefficient for pure refrigerant. It is seen that there is an increase in the local heat transfer coefficient in the beginning of the evaporator even when the mean value is lower with oil than without. There is an increase in oil content in the liquid along the evaporator since more and more refrigerant appears in gaseous phase. In the beginning of the evaporator, the oil might work as a nucleation help, whereas the very oil rich liquid at the end of the evaporator probably works as an insulator, thus the decrease of the heat transfer coefficient.

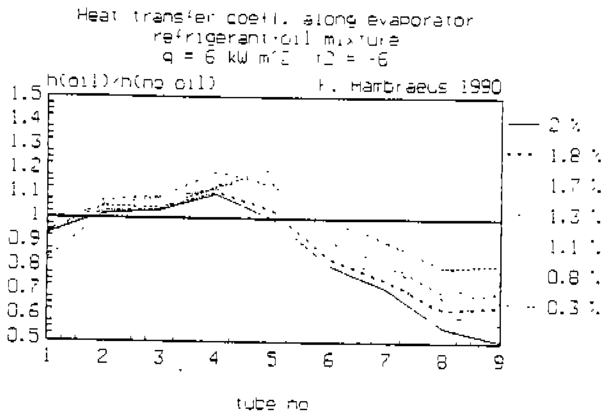


figure 10

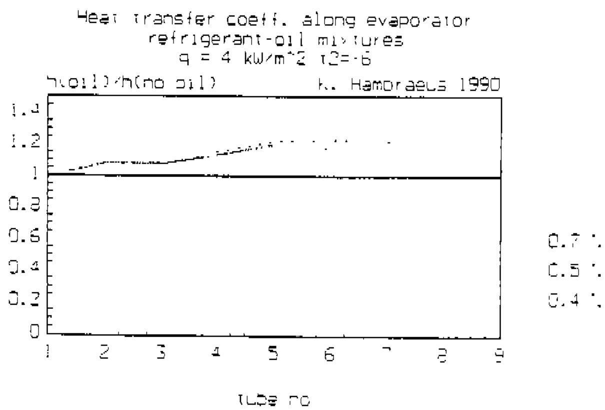


figure 11

DISCUSSION

Refrigerant temperature in the evaporator is calculated with the help of measured pressure. If the relation between pressure and temperature (vapor pressure curve according to Wilson-Basu) is wrong, this will result in erroneous heat transfer coefficients ($h=q/\Delta t$). It is unlikely though, that the great difference in heat transfer coefficient compared to HCFC22 should be due to errors in vapor pressure curve. The deviations from dimensionless correlations, however, are probably mainly due to uncertainties in the properties of HFC134a. A more detailed study is needed to check the influence of properties and other parameters as for example Reynolds number. It would also be interesting to know the influence of oil on refrigerant properties, such as vapor pressure curve and viscocity.

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