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HEAT TRANSFER COEFFICIENTS OF HFC REFRIGERANTS DURING CONDENSATION AT HIGH TEMPERATURE INSIDE AN ENHANCED TUBE

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ABSTRACT

Experimental heat transfer coefficients during condensation of R134a and R407C in a microfin tube are reported. Heat transfer measurements are compared against performance of an equivalent smooth tube under the same operating conditions, to show advantages of the microfin tube as compared to the smooth tube. Experimental tests are carried out in a broad range of operating temperatures to enlighten the influence of saturation temperature on the heat transfer coefficient. Comparisons with three models available in the literature are reported for the two fluids. For the zeotropic mixture R407C, a suitable correction is used in the heat transfer prediction procedure valid for pure fluids.

NOMENCLATURE

A: surface area [m^2]
α: heat transfer coefficient [$W/(m^2 K)$]
c_p: specific heat [J/(kg K)]
ΔT_{GL}: temperature glide [K]
G: mass velocity [$kg/(m^2 s)$]
h: enthalpy [J/kg]
h_{LG}: latent heat [J/kg]
q: heat flow rate [W]
T: temperature [K]
T_{dew}: dew point temperature [K]
x: vapour mass quality

INTRODUCTION

Since the end of 1970s, condensation heat transfer inside horizontal tubes in heat exchangers for air conditioners is enhanced by finned tubes. During condensation, microfin tubes show a heat transfer enhancement when compared with equivalent smooth tubes under the same operating conditions, that is partly due to the mere increase in the effective exchange area, and additionally to the turbulence induced in the liquid film by the micro fins and to the surface tension effect on the liquid drainage.

Although the condensation process inside microfin tubes usually takes place at high saturation temperature (50-60°C), as is the case of air-cooled condensers, most of the experimental heat transfer coefficients reported in the literature are taken at lower temperature (around 40°C), which is more typical for water-cooled chillers.

The performance of a microfin tube during condensation has been studied experimentally at 40°C and 55°C in this work. The authors present their own data when condensing R134a and R407C inside a 9.5 mm outer diameter microfin tube at mass velocities ranging from 100 to 800 $kg/(m^2 s)$ by direct measurement of the wall temperature. The test tube has a 7.69 mm inside diameter at the base of the grooves and 60 fins with 0.23 mm fin height and 13° helix angle. Fins have trapezoidal shape with smoothed tip and 43° apex angle. An enlarged image of the fins is reported in Figure 1.

R134a is a HFC pure refrigerant, while R407C is a zeotropic mixture of HFC-32/125/134a (23/25/52% by mass). R407C has recently been employed as a short term alternative working fluid for R22 in air-conditioning equipment. It has been found that R407C brings some heat transfer problems in the practical use of air-conditioning units, namely the degradation of heat transfer coefficients due to the mass transfer thermal resistance build-up.

Microfin tubes are used in air-conditioners operating with R407C, and therefore it is really important to know the performance of this mixture in respect with this particular geometry. A comparison among heat transfer coefficients measured inside the same microfin tube for R22 and R407C is given in Cavallini et al. (2002b).

Measured heat transfer coefficients at high condensing temperature can be compared against semi-empirical predictive correlations for condensation heat transfer coefficients. In this way, such correlations can be checked and critically reviewed under an extended range of operating conditions, particularly with regard to their capability to predict heat transfer in a broad temperature range.

EXPERIMENTAL PROCEDURE

Test facility

The experimental tests are run in a test section set up at the Dipartimento di Fisica Tecnica of the University of Padova. A schematic of the apparatus is reported in (Cavallini et al., 2001). The experimental facility consists of three loops: the refrigerant loop, the cooling water loop and the hot water loop. In the primary loop the refrigerant is vapourised and superheated in two tube-in-tube heat exchangers, heated by hot water. Then it partially condenses in the precondenser to achieve the set quality at the inlet of the test section.

The test section is a counter flow tube-in-tube condenser, with the refrigerant condensing inside the inner tube, against the cold water flowing in the annulus. The measuring heat transfer section is 300 mm long and it is instrumented with thermocouples embedded in the tube wall in the middle of the tube: it has four thermocouples, soldered circumferencially to draw a cross shape.

Refrigerant temperatures at inlet and outlet of the test section are measured by means of adiabatic sections, using thermocouples inserted into both the refrigerant flow and the tube wall. The refrigerant flow can be independently controlled by a magnetically coupled gear pump. Two digital strain gauge pressure (absolute and differential) transducers are connected to manometric taps to measure the vapour pressure upstream and downstream of the test tube. The refrigerant mass flow rate is measured by a Coriolis effect mass flow meter inserted downstream of the pump. The cooling water flow rate is measured by a magnetic flow meter and its temperature gain across the instrumented test tube is measured with a differential four-junction copper-constantan thermopile, installed into mixing chambers to assure perfect mixing of the water. The actual composition of the mixture flowing inside the test rig is measured by in-line gas-chromatography.

The average coolant flow rate during tests is around 250 l/h and the saturation-to-surface temperature difference keeps between 4°C and 12°C. It was estimated, from a propagation of error analysis, that the heat transfer was measured to an accuracy of ±4% and the heat transfer coefficient to an accuracy of ±4.5% at typical test conditions, with maximum uncertainty of ±7.5% at worst conditions for the heat transfer coefficient. A list of accuracies for sensors and parameters is reported in Table 1.

Temperature	± 0.05°C	Absolute pressure	± 7 kPa
Temperature difference	± 0.03°C	Vapour quality	± 0.04
Water flow rate	± 0.2%	Heat flow rate	± 4.0%
Refrigerant flow rate	± 0.2%	Heat transfer coefficient	± 4.5%

Table 1: Accuracy for sensors and parameters (at typical test conditions)

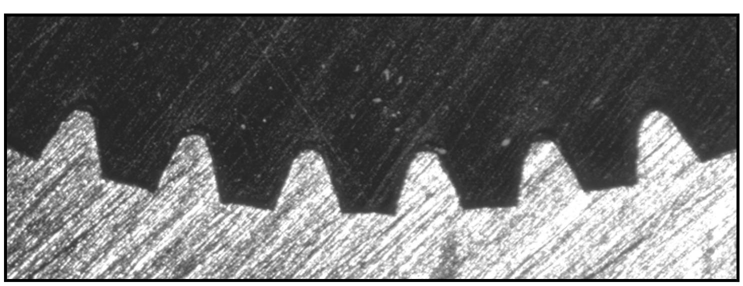

Figure 1: Enlarged image of the fins.

Data reduction

The heat flow rate transferred in the test tube is derived from a thermal balance on the cooling water side. The average condensation heat transfer coefficient for R134a and R407C is obtained as:

$$\alpha = q / (A\ \Delta T)$$

where q is the heat flow rate exchanged in the tube, A is the transfer surface area of a smooth tube with the same diameter as the fin tip diameter of the microfin tube and ΔT is the temperature difference between the vapour and the tube wall.

The vapour quality entering the test section (x_{in}) is calculated from an energy balance on the precondenser. For a pure refrigerant or an azeotropic mixture, the vapour quality change is given as a ratio of the isobaric change in enthalpy in the test section δh to the latent heat h_{LG}.

Condensation of the refrigerant blend R407C differs from that of a pure refrigerant, such as R134a, in that the isobaric process takes place over a temperature range or glide for the mixture rather than at a fixed saturation temperature as for the pure refrigerant. Hence, the heat removed from the refrigerant blend not only includes latent heat from the phase change process but also sensible cooling of the vapour and liquid phases as they are cooled to a lower temperature. The isobaric change in enthalpy, δh, of a mixture during condensation along a tube can be given as

$$\delta h = h_{LG}\ \delta x + (1-x)\ (c_p)_L\ \delta T_{dew} + x\ (c_p)_G\ \delta T_{dew}$$

The above equation reduces to only the latent heat term when applied to a pure fluid or an azeotropic mixture. The values of h_{LG} , $(c_p)_L$ and $(c_p)_G$ depend on saturation temperature, as is the case for pure refrigerants. But they are also a function of the local liquid and vapour compositions.

The change in temperature δT_{dew} is determined from the following expression:

$$\delta T_{dew} = \Delta T_{GL}\ [\ f(x) - f(x - \delta x)\ ]$$

where the temperature glide ΔT_{GL} is defined as the difference between the dew point temperature and the bubble point temperature at a fixed pressure with the same composition in the vapour phase as in the liquid phase, while f(x) is an empirical function that varies from 0 to 1.0. For approximate calculations, the value of f(x) can be determined from a linear variation f(x) = x. For reduction of the present experimental data, the relationship between f(x) and x reported in (Zurcher et al., 1998) was applied.

Condensation tests are carried out in the same microfin tube at around 100, 200, 400 and 800 kg/(m^2s) mass velocities. The average inlet vapour quality varies between 0.2 and 0.8, and approximately 10 – 25% vapour quality change occurs in the test tube depending on the mass flux velocity and heat flux. The heat transfer values presented are actually mean values over a small change of vapour quality and can be referred to as quasi-local values.

EXPERIMENTAL RESULTS

Heat transfer coefficients at constant saturation temperature

Figure 2 shows the condensation heat transfer coefficient for R134a, referred to the surface area of a plain tube with 7.69 mm internal diameter, plotted as a function of the vapour quality at 40°C saturation temperature.

It can be seen that at high mass velocity, 800 kg/(m^2 s), the heat transfer coefficient varies linearly with vapour quality, as observed during condensation in smooth tubes (Cavallini et al., 2001). At lower mass velocity, around 200 kg/(m^2 s), the effect of the fins may explain the non-linear trend reported in Figure 2, which is not observed in a smooth tube. With respect to the linear trend, the heat transfer coefficient increases at high vapour quality while decreases at low vapour quality.

The heat transfer enhancement factor is plotted vs. vapour quality in Figure 3, where the enhancement factor is defined as the ratio of heat transfer coefficient in the microfin tube to the heat transfer coefficient in a equivalent ID smooth tube at the same operating conditions. The internal diameter at the fin tip is taken as reference in the microfin tube. Heat transfer coefficients for the reference smooth tube were calculated by the new model by Cavallini et al. (2002a). This model is a semiempirical correlation that was obtained from a best fitting procedure on data measured by the same authors (Cavallini et al., 2001) in a 8 mm internal diameter smooth tube.

The heat transfer enhancement factor depends on mass velocity and vapour quality: the maximum value of the enhancement factor for R134a at 40°C saturation temperature is obtained at 200 kg/(m^2s) mass velocity, where it can reach up to 2.8. As the area enhancement for the test microfin tube is equal to 1.8 ($A_{MICROFIN}/A_{SMOOTH}$ = 1.80), it can be seen that heat transfer enhancement is not merely due to the area enhancement, and other effects are important to be accounted for. For values of mass velocity higher than 200 kg/(m^2 s), the enhancement factor decreases when mass velocity increases. At 800 kg/(m^{2}s) the heat transfer coefficient is around 50% higher as compared with the smooth tube, that is the heat transfer enhancement is lower than the area enhancement. These same results were also found by present authors during tests with R22 (Cavallini et al., 2002b).

Similarly, for G=100 kg/(m^{2}s) the enhancement factor is lower as compared to the case at 200 kg/(m^{2}s). It comes out that there is an optimal value of mass velocity with respect to the heat transfer performance of a microfin tube. This may be due to the flow pattern in the tube. In fact, at low values of mass velocity, microfins could promote the annular flow pattern, leading the maximum enhancement for a certain value of the mass velocity.

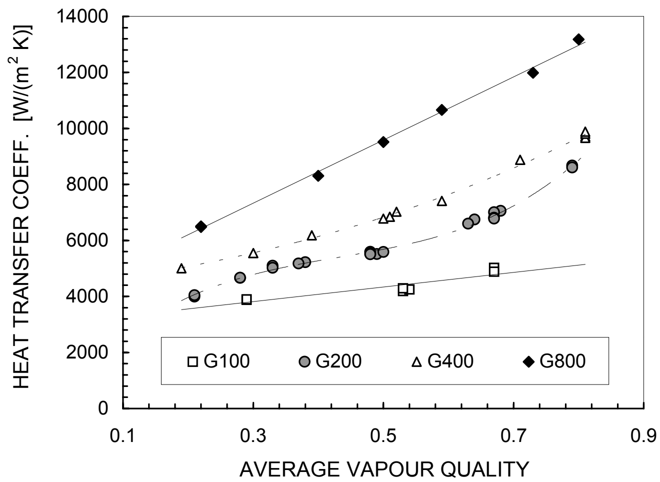

Figure 2: Heat transfer coefficients for R134a at 40°C saturation temperature. G is mass velocity [kg/(m^{2}s)].

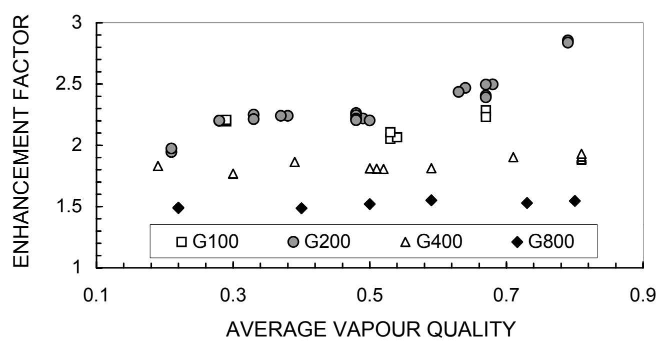

Figure 3: Heat transfer enhancement factor for R134a at 40°C saturation temperature.

The heat transfer coefficient measured during condensation of R134a at 55°C saturation temperature in the same microfin tube is reported in Figure 4, while the heat transfer enhancement factor at the same temperature is shown in Figure 5. As it can be seen from Figure 3 and Figure 5, at 400 and 800 kg/(m^2 s) mass velocity the enhancement factor is roughly the same at the two temperatures 40°C and 55°C, and only at 200 kg/(m^2 s) and high values of vapour quality a difference can be observed between the two temperatures with regard to the enhancement factor. Anyway, a larger number of points taken at 55°C would be necessary to draw a definitive conclusion about the effect of saturation temperature on the heat transfer enhancement at low mass velocity (100 and 200 kg/(m^2 s)) and high vapour quality.

The heat transfer coefficient measured when R407C condenses at an average pressure of 23.5 bar and at mass velocity between 100 and 800 kg/(m^2 s) is plotted in Figure 6 (G is mass velocity).

When the heat transfer coefficients of R134a (Figure 4) and R407C (Figure 6) are compared at the same operating conditions (saturation temperature, mass velocity and vapour quality), it appears evident that the difference between the coefficients of the two fluids depends on the flow regime. In fact R407C heat transfer coefficients are significantly lower (around 25%) as compared with those of R134a at 200 kg/(m^{2}s), but the difference is around 15% at 800 kg/(m^{2}s) mass velocity. This result suggests that the effects of the zeotropic characteristics of the mixture depend on the mass velocity.

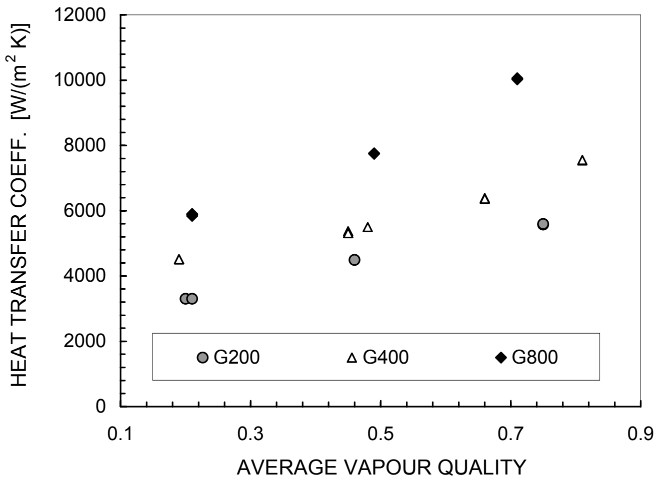

Figure 4: Heat transfer coefficients for R134a at 55°C saturation temperature. G is mass velocity [kg/(m^{2}s)].

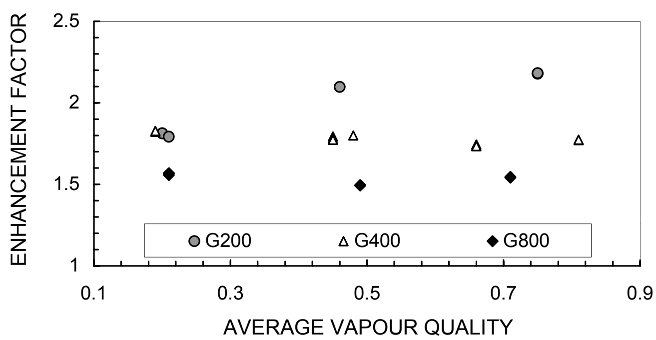

Figure 5: Heat transfer enhancement factor for R134a at 55°C saturation temperature..

Heat transfer coefficients when varying saturation temperature

Experimental tests were also carried out during condensation of R134a at a wider range of saturation temperatures: local heat transfer coefficient for 400 kg/(m^{2}s) mass velocity are plotted vs. temperature in Figure 5. It can be seen that at 0.5 vapour quality the heat transfer coefficient drops down from 7900 W/(m^{2}K) at 30°C to 5060 W/(m^{2}K) at 60°C, with a 36% reduction. At higher saturation temperature, lower heat transfer coefficients correspond to lower pressure drop.

R134a heat transfer coefficients are plotted vs mass velocity at two values of saturation temperatures (40°C and 55°C) and constant vapour quality in Figure 6. As it can be expected, the experimental values measured at 40°C are always higher when compared to the values at 55°C. Experimental values are also compared against predictive procedures by Cavallini et al. (1999), Kedzierski and Goncalves (1997) and Yu & Koyama (1998).

The model by Cavallini et al., that was developed from independent research databases, shows to predict satisfactorily experimental data at 40°C and to be able to reproduce the experimental trend of heat transfer coefficient vs. mass velocity. Although it gives the lowest deviation with experimental data at 40°C, it does not properly account for the decrease of the heat transfer coefficient at higher saturation temperature. This is probably due to the narrow range of operating temperature values of the data bank used for developing the model.

The correlation by Yu and Koyama shows to be able to predict the experimental heat transfer coefficients only at 200 kg/(m^{2}s), while it overpredicts experimental values at higher mass velocity both at 40 and 55°C. Anyway, since this equation was developed from the Haraguchi et al. (1994) model for smooth tubes, it should be noted that experimental data points at mass velocities higher than 400 kg/(m^2 s) do not fall inside the validity range of the Haraguchi et al. model.

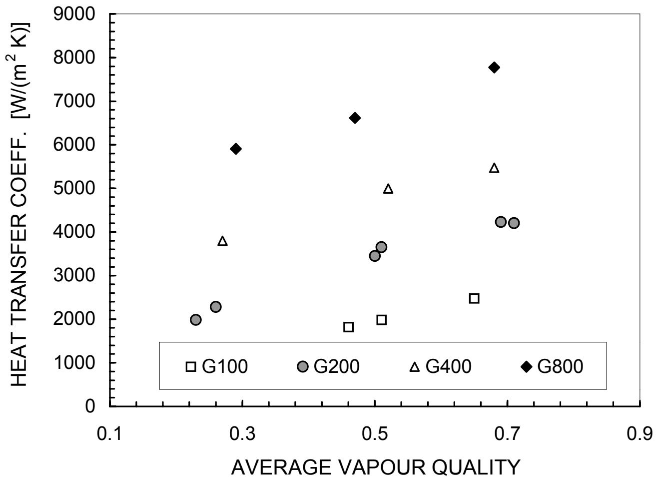

Figure 6: Heat transfer coefficients for R407C at 55°C average saturation temperature.

The third model, by Kedzierski and Goncalves (1997), obtained by regression of their own data, better accounts for the influence of saturation temperature on the heat transfer coefficient.

For R407C, due to the mass diffusion resistance build-up, it is suggested to calculate the heat transfer coefficient by applying the correction by Bell and Ghaly (1973) along with the pure refrigerant models already mentioned. This way, the mixture heat transfer coefficient is calculated as

$$\alpha_m = [1/\alpha + x\ c_{pG}\ (\Delta T_{GL}/h_{LG})\ /\ \alpha_G]^{-1}$$

where α_m is referred to the temperature difference between saturation (varying between dew point and bubble point) at the operating pressure and tube wall, α is the condensate film heat transfer coefficient computed by one of the above models for pure fluids, α_G is the heat transfer coefficient of the vapour phase flowing alone in the duct and calculated by the classical Dittus-Boelter equation.

Experimental and calculated values of the heat transfer coefficient for the mixture R407C at 0.5 vapour quality are plotted vs. mass velocity in Figure 7, both at 40°C and 55°C. The three models applied in the case of R134a are also used here to calculate the heat transfer coefficient, which is then corrected by the Bell and Ghaly correction as already explained. As for the case of R134a, the model by Yu and Koyama overpredicts the heat transfer coefficient at 800 kg/(m^2 s). Besides this correlation seems not to predict satisfactorily the experimental trend of heat transfer coefficient vs. mass velocity. A better agreement is obtained by applying the model of Cavallini et al. (1999) with the Bell and Ghaly correction to data at 40°C saturation temperature. As in the previous case, this model does not follow the decrease of the coefficient when temperature increases. At 100 kg/(m^{2}s) the model by Cavallini et al. is applied even if operating conditions fall outside validity ranges given by the authors.

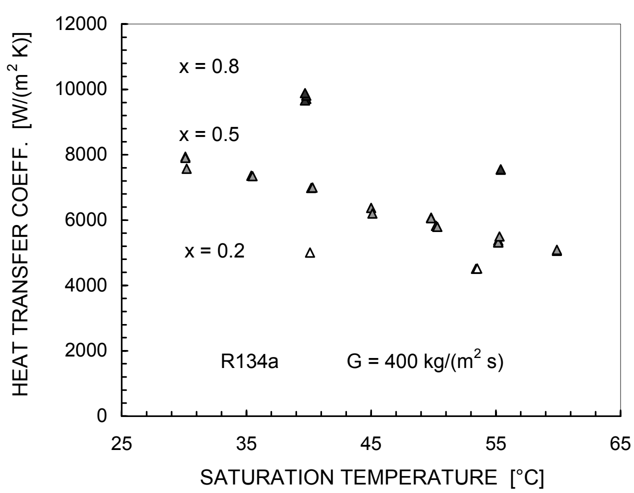

Figure 7: Experimental heat transfer coefficients for R134a at varying saturation temperature.

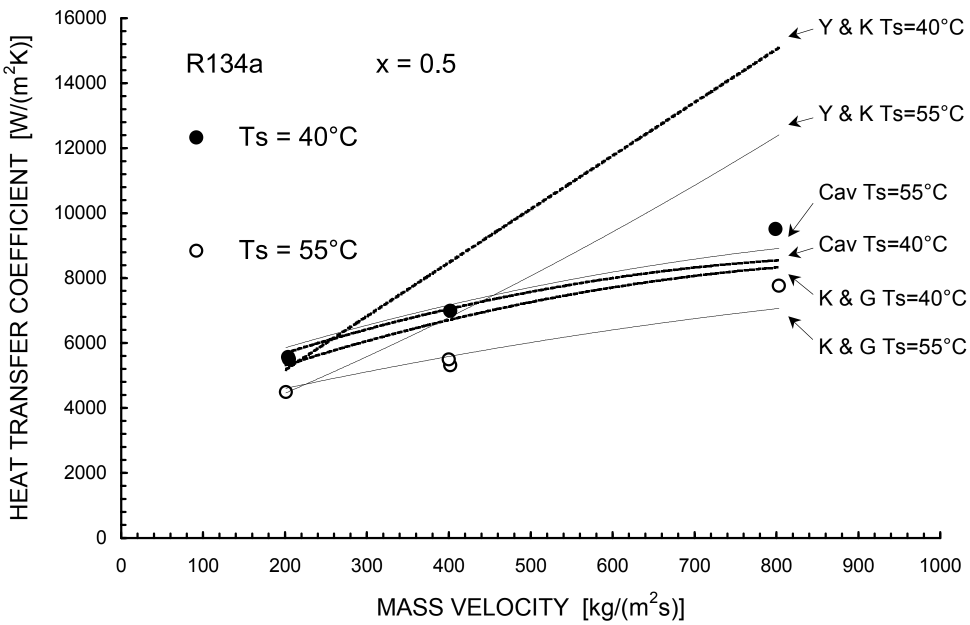

Figure 8: Experimental (markers) and calculated (lines) heat transfer coefficients for R134a at 40°C and 55°C saturation temperature. The following models are applied: Cavallini et al. (1999) [Cav], Kedzierski and Goncalves (1997) [K & G] and Yu & Koyama (1998) [Y & K].

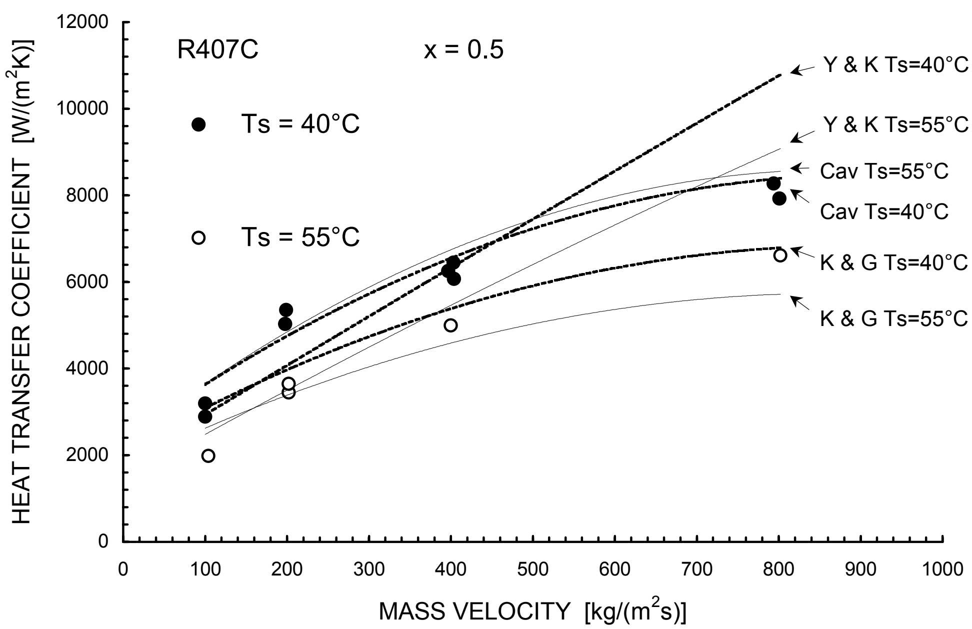

Figure 9: Experimental (markers) and calculated (lines) heat transfer coefficients for R407C at 40°C and 55°C saturation temperature. The following models are applied: Cavallini et al. (1999) [Cav], Kedzierski and Goncalves (1997) [K & G] and Yu & Koyama (1998) [Y & K]. All these three models are applied along with the Bell and Ghaly correction (1973).

The model by Kedzierski and Goncalves is also applied to the case of R407C showing a general underprediction of the experimental values. The disagreement between calculated and experimental values in this case increases with increasing mass velocity.

As already stated, the comparisons reported in Figure 6 and Figure 7 enlightens only the agreement between models and experimental data at the average value of vapour quality 0.5. Comparisons among calculated and experimental values of heat transfer coefficients when varying vapour quality are reported in Cavallini et al. (2002b).

CONCLUSIONS

In-tube condensation tests for R134a and R407C are reported for a microfin tube. The experimental results related to R134a local heat transfer coefficients condensing in the microfin tube at 400 kg/(m^2s) show a 36% reduction of the coefficient when temperature decreases from 30°C to 60°C. This result has to be accounted for when designing a condenser using microfin tubes, because its performance can be dramatically different in the case of a water-cooled condenser (lower saturation temperature), from the case of an air-cooled condenser (higher saturation temperature).

R134a heat transfer measurements are also compared against performance of a smooth tube under the same operating conditions. The heat transfer enhancement factor for R134a depends very much on the mass velocity in the tube: it reaches 2.8 at 200 kg/(m^2s) mass velocity and 0.8 vapour quality. At high mass velocity (800 kg/(m^2s)), the heat transfer enhancement does not depend on vapour quality and its value is lower than the surface area enhancement.

The heat transfer performance of the mixture R407C is also affected by the mass transfer resistance during the condensation process. Both the heat transfer enhancement due to the grooves and the degradation of the heat transfer coefficient due to the zeotropic characteristics of the mixture depend on the mass velocity. It comes out that the heat transfer coefficient of R407C is lower than that of R134a inside a microfin tube at the same operating conditions, and this underperformance increases when mass velocity decreases.

Comparisons with the models available in the literature are reported for R134a and R407C data. It was shown that the model by Cavallini et al. gives the best agreement with experimental data at 40°C saturation temperature while the Kedzierski and Goncalves correlation reproduces better the decrease of heat transfer coefficient with increasing saturation temperature.

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