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CONDENSATION OF R134a INSIDE A NEW MICROFIN TUBE

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ABSTRACT

This paper presents experimental data on heat transfer coefficients and pressure drops during convective condensation of oil-free R134a inside a 9.52-mm microfin tube of new design and in a smooth tube with the same outside diameter. Data previously obtained for R22 in the same microfin tube are also reported for comparison, although R134a is a R22-substitute only in some applications. All the tubes are horizontally operated. The microfin tube is characterized by 54 sharp fins alternating with two different heights, that is the distinguishing feature from other new microfin tubes. The effect of mass flux, average quality and overall quality change on heat transfer characteristics have been separately investigated. The majority of data here reported are for a nominal temperature of 35 °C whereas some series are for 25 °C, with mass flux ranging from 90 to 400 kg/ s m^2, inlet quality from 0.8 to 0.4 and quality change between 0.6 and 0.2. Finally, experimental data on heat transfer and pressure drop are compared to predictions of some recent correlations specifically proposed for condensation of conventional and alternative refrigerants inside microfin tubes.

INTRODUCTION

The design of high performance compact heat exchanger led to develop many types of extended surfaces. In refrigeration and air-conditioning applications, reduction of the air-side thermal resistance to the refrigerant-side values urged industry to focus on enhancing heat transfer during in-tube evaporation and condensation. Among enhancement techniques, microfin tubes are widely used because of their high heat transfer performance with moderate pressure drop increase. Microfin tubes are characterized by numerous small fins that spiral down the inside surface, and they were first reported in the open literature in the mid 1970's. In the subsequent years, an increasing number of additional papers on boiling and condensation of refrigerants inside these tubes have been published. Comprehensive reviews of the previous works are reported by Schlager et al . [1,2] and Webb [3].

More recently, international regulations banned the use of CFCs like R12 and imposed phase-out schedules on the production of HCFCs like R22. Consequently, an extensive search for potential replacements has been made during the past decade. R134a was the first alternative refrigerant and may be designated today as state-of-art; in its physical and refrigeration properties, it compares very well to R12. In contrast to the situation for R12, there is not a prevailing substitute of R22 because no single pure or blended substance can offer all the technical advantages of R22. For instance, for air-conditioning applications with evaporating temperatures approximately of 0 °C, R134a yields a coefficient of performance higher than that of R22 but it is characterized by a smaller refrigerating effect per unit of swept volume. On the contrary, R410A allows a refrigerating effect up to 50% above that of R22 but it can not be used as a retrofit for existing R22-systems because of its higher working pressure. Finally, R407C was developed to reproduce the vapor pressure curve of R22; however, this refrigerant is a zeotropic mixture and its temperature glide can lead to problems in case of leakage.

The appearance of refrigerant substitutes redirected research on microfin tubes towards testing their heat transfer characteristics with these new fluids. References [4,10] are a sample of the many papers recently published on these investigations. However, evaporation and condensation of alternative refrigerants inside microfin tubes have not been thoroughly surveyed yet and there is a large demand for further experimental research.

This paper presents experimental data on convective condensation of R134a in a microfin tube of new design developed by Trefimetaux, and in a smooth tube. Although R134a can be a R22-substitute only in some applications as aforementioned, the paper also reports comparisons with data previously obtained by the authors during condensation of R22 inside the same microfin tube. Finally, experimental data are compared to predictions of some recent correlations specifically proposed for condensation of conventional and alternative refrigerants inside microfin tubes.

EXPERIMENTAL APPARATUS

A schematic diagram of the experimental facility is shown in Figure 1. The apparatus is composed of three independent circuits, namely, a refrigerant circuit, a heating/cooling water circuit and a chilled coolant (water-glycol solution) circuit.

The refrigerant circuit mainly consists of a boiler, the test section, a condenser, a gear pump and a filter dryer. Liquid and vapor are drawn from the boiler through two distinct lines. A subcooler and a mass flow-meter are mounted on the liquid line whereas a superheater and two float-type flow-meters are installed on the other line. Subcooler and superheather ensure

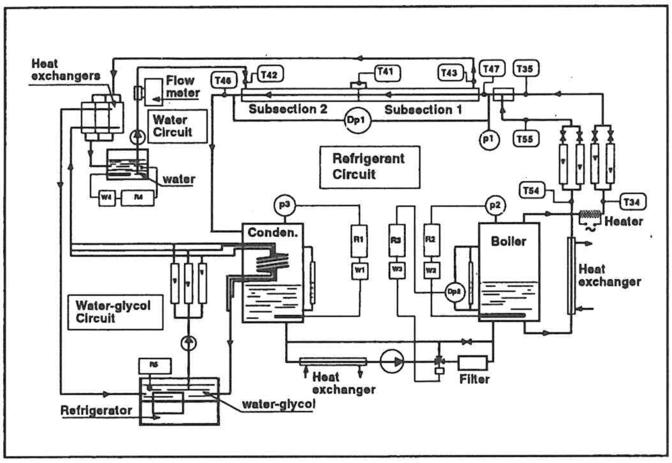

Figure 1. Schematic diagram of the experimental facility.

a single-phase flow through the flow-meters for any operating conditions. The liquid and vapor flow rates are controlled by precision metering valves. Downstream of the valves, vapor and liquid streams are mixed; the resulting two-phase mixture flows through a 1.5 m long calming section and then it enters the test tube, that is the inner tube of a double-pipe heat exchanger. At the exit, the refrigerant is discharged into a shell-and-coil condenser; a gear pump conveys the refrigerant from the condenser to the boiler. The heating/cooling circuit supplies the water flowing in the annulus outside the test tube. This circuit is composed by a centrifugal pump, a plate heat exchanger and a bath vessel equipped with a 5-kW heater controlling the water temperature. A magnetic flow-meter is used to measure the water flow rate through the test section annulus. Finally, the chilled coolant circuit is filled with a water-glycol solution and it supplies the cold medium circulating into heat exchangers mounted on the refrigerant and water circuits. A commercial refrigeration unit is used as chiller.

The test section is divided into two identical subsections mounted in series; each subsection is 1.3-m long with an effective heat transfer length of 1.12 m, that is the distance between the inlet and outlet ducts of the outside tube. At the inlet of test section, a temperature probe is mounted inside the tube countercurrently to the refrigerant flow. Such a probe consists of a 0.25-mm, K-type thermocouple plugged into a L-shaped, 8-cm long, 2-mm o.d. tube that is located at the centerline of the duct. In addition, at the entrance and exit of the first subsection and at the outlet of the second one, there are two pressure-taps. Downstream the exit pressure taps of each subsection there is a sight glass made of 85-mm long, 8.5-mm i.d., pyrex smooth tube; sight glasses are neither heated or cooled. Finally, each subsection is equipped with four T-type thermocouples to measure wall temperatures; the thermocouples are placed in pairs on the top and the bottom of the test tube. Each thermocouple is cemented in a longitudinal groove cut in the outside wall of the test tube, with the tip at 140 mm from the tube end and at 50 mm from the inlet/outlet duct of the outside annulus. Calming and test sections are thermally insulated by a 10-cm thick, glass-wool covering that ensures a measured thermal resistance of 4 K/W.

DATA REDUCTION

Signals from thermocouples and transducers are cyclically read by a data acquisition unit and sent to an on-line PC. In order for all variables to be affected by similar RMS relative errors, the measurements of refrigerant temperature, pressure drop and water flow rate are based on 30, 50 and 100 readings for cycle, respectively. Every experimental value, instead, is obtained by averaging the measurements of ten cycles in order to reduce the influence of random errors and fluctuations. Finally, for every operative condition, more than ten experimental data are collected.

The heat transfer coefficient is computed as follows. We assume the refrigerant temperature varies linearly between the value T_{in}, measured at the entrance of the test section, and the value T_{out} computed at the exit as $T_s(p_s(T_{in}) - \Delta p)$, where T_s is the function correlating the saturation temperature to the pressure, p_s the inverse function of T_s, and Δp the pressure drop measured along the test section. Then, for each subsection we calculate the mean refrigerant temperature $T_{r.m,i}$, the mean wall temperature $T_{w.m,i}$, the refrigerant to wall temperature mean difference $\Delta T_{m,i}=(T_{w.m,i}-T_{r.m,i})$, and the heat transfer coefficient $h_i=q_i/\Delta T_{m,i}$ where q_i is the mean heat flux based on a nominal inside area corresponding to the maximum internal diameter, i.e., the diameter at the root of microfins. Eventually, we compute the average heat transfer coefficient for the test section as the arithmetic mean of the subsection coefficients h_i. Relevant variables for the present investigation are affected by the following representative experimental uncertainties measured or estimated by an error propagation analysis: ±2.8% for the refrigerant mass flow rate, ±1.3% for the inlet quality, ±0.2 K between the refrigerant temperature and the saturation one, ±1.0% for the refrigerant pressure drop, ±1.0% for the water volume flow rate, ±0.02 K for the water temperature difference between the subsection inlet and outlet, ±1.4% for the heat rate, and ±7% for the average heat transfer coefficient.

RESULTS AND DISCUSSION

In saturated flow boiling, for a fixed section configuration (shape, dimensions and orientation with respect to gravity), average heat transfer coefficient and pressure drop depend on four independent variables, namely, total mass flow rate, temperature (or pressure), inlet thermodynamic quality and heat rate or quality change over the section, being the fluid everywhere saturated.

The experimental data here reported were obtained with oil-free refrigerant R134a inside a 9.52-mm microfin tube of new design and in a smooth tube with the same outside diameter. Data previously obtained with R22 are also reported for comparison. The microfin tube was developed and manufactured by Trefimetaux, namely, Metofin 952-30VA40/54, and it is characterized by 54 sharp fins with an apex angle of 40 degrees, alternating with two different heights as shown in Figure 2. The latter is the distinguishing feature from other new microfin tubes. Table 1 lists geometrical parameters of the microfin and smooth tubes, as well as the heat transfer internal surface ratio and the actual cross-section ratio with respect to

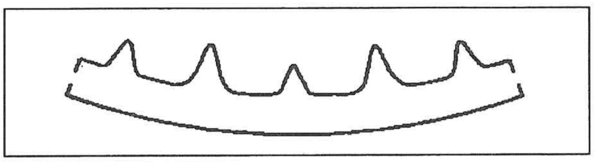

Figure 2. Drawing of the microfin cross-section profile.

Parameter	Tube	VA	Smooth
Outside diameter	[mm]	9.52	9.52
Maximum inside diameter	[mm]	8.92	8.92
Bottom wall thickness	[mm]	0.30	0.30
Higher fin height	[mm]	0.23	-
Lower fin height	[mm]	0.16	-
Apex angle		40°	-
Number of grooves		54	-
Helix angle		18°	-
Heat transfer internal surface ratio		1.58	1
Actual cross-section surface ratio		0.963	1

Table 1. Geometrical parameters of the tested tubes.

the smooth tube. The majority of tests were carried out at a nominal temperature of 308 ± 0.2 K, corresponding to a pressure of 0.35 MPa for R134a and 0.58 MPa for R22; some series of data are for a nominal temperature of 298 ± 0.2 K. Total mass flow rate, inlet thermodynamic quality and quality change were varied in turn while keeping the others constant in order to show clearly the effect of each variable against the others. The total mass flow rate ranged from 5.56 to 25 g/s corresponding to a mass flux G, with respect to a nominal cross-section area based on the maximum internal diameter, that varies between about 90 and 400 kg/s m^2. The inlet quality x_{in} was varied from 0.8 to 0.4 whereas the quality change x ranges from 0.2 to 0.6.

Figure 3 displays the average heat transfer coefficient h_c plotted versus the mass flux G for the microfin and smooth tubes; data are for inlet quality x_{in}=0.7 and quality change Δx=0.4. In considering this figure, it is worth to note that mass flux variations at constant Δx imply proportional variations in heat flux due to their linear dependence; for the data reported in figure, the average heat flux ranges from 5.2 to 24 kW/m^2. As expected, the heat transfer coefficient is an increasing function of G, but the trend for the microfin tube differs from the smooth tube. For the latter, data exhibit a linear-at-interval dependence on G with a change of slope approximately at G=250 kg/ s m^2 for both refrigerants. Supported by visual observations, we infer that in the first region, where heat transfer is weakly dependent on the mass flux, the flow is stratified whereas it is annular when h_c starts to increase more steeply with G. For the microfin tube, instead, data do not display any change of slope marking the transition from stratified to annular flow. This trend seems to suggest that microfins enhance heat transfer more effectively at low values of mass flux when the flow is stratified than at high G in annular flow. We conjecture that in stratified flow microgrooves hold up the condensate at the top of the tube and they promote the formation of a thicker liquid film which in turn lowers the liquid pool. As well known, a redistribution of condensate from the pool to the film yields a higher heat transfer coefficient. On the other hand, in the annular flow microgrooves spread condensate over all the inside surface of the tube thinning the liquid film and increasing heat transfer; however the ratio of the heat transfer coefficient to that of a smooth tube, i.e., the enhancement factor, is lower than in stratified flow and decreases as mass flux increases. Regarding the working fluids, the values of h_c for R134a inside the smooth tube are greater than those of R22, whereas in the microfin tube the refrigerant R134a gives values quite lower than those of R22 (it is worth, however, warning that data for R22 were collected before of those for R134a and that meanwhile the VA-tube could have undergone to fouling effects).

Differences in thermal performances are well accounted by the enhancement factor. A plot of this factor versus G is shown in Figure 4. As seen, for R22 the enhancement factor exhibits a slightly marked maximum of 2.1 (which comes up to 2.6 for the data, not shown, obtained at x_{in}=0.8 and Δx=0.6) and thereafter decreases with G and tends to about 1.8 that is a value higher than the internal-surface-ratio. This result disagrees with the conclusion by Eckels and Pate [9] that at high mass flux the heat transfer augmentation is due to the area increase only, whereas geometry seems to affect fluid dynamics and heat transfer even at high G, as observed by Ito and Kimura [10], too. For R134a the enhancement factor is a weakly increasing function of the mass flux up to G=266 kg/s m^2; at this value it comes up to a maximum of 1.5 and thereafter decreases to 1.25 that is lower than the internal-surface-ratio. Consequently, the enhancement factor with R134a is markedly lower than that with R22. Results qualitatively similar to those

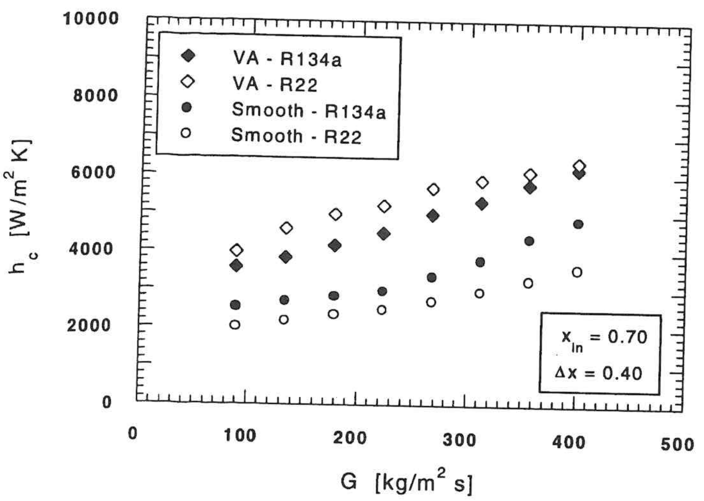

Figure 3. Heat transfer coefficient h_c versus mass flux G for fixed inlet quality x_{in} and quality change Δx.

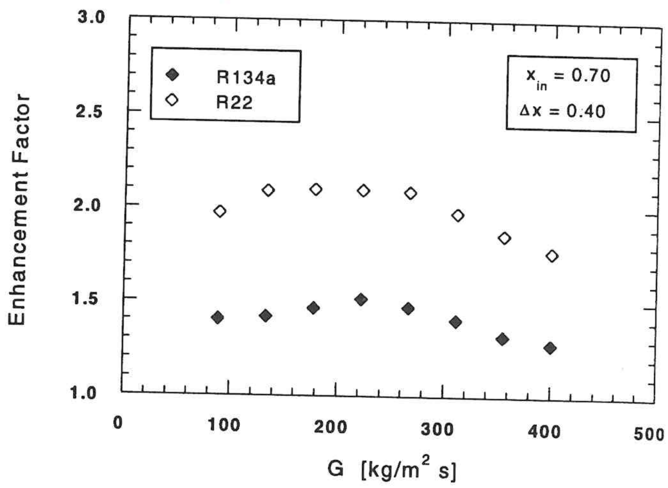

Figure 4. Enhancement Factor versus mass flux G for fixed inlet quality x_{in} and quality change Δx.

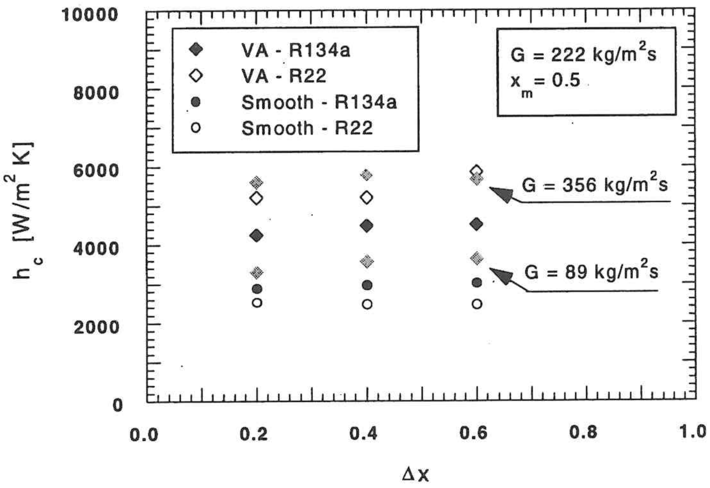

Figure 5. Heat transfer coefficient h_c versus quality change Δx for fixed mass flux G and inlet quality x_{in} .

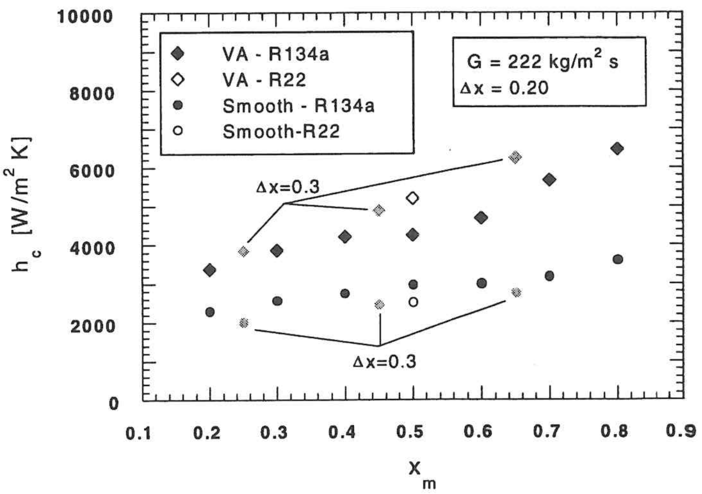

Figure 6. Heat transfer coefficient h_c versus average quality x_m for fixed mass flux G and quality change Δx.

reported in Figure 3 have been obtained for quality changes of 0.2 and 0.6 while keeping fixed the average quality. Figure 5 shows the dependence of h_c on the quality change Δx at x_m=0.5 for three different values of G, i.e, 89, 222 and 356 kg/s m^2. As seen, the trends are essentially flat and therefore h_c is quite independent on Δx.

Finally, Figure 6 shows the influence of the average quality x_m on h_c at G=222 kg/s m^2 and Δx=0.2. Results for R22 include indeed data for Δx=0.3; this discrepancy is quite irrelevant because of the weak influence of Δx on h_c. For the smooth tube, the heat transfer coefficient is an almost linear increasing function of the average quality with values for R134a slightly higher than for R22. For the microfin tube, h_c is also an almost linear function of x_m for both refrigerants, but the data for R134a exhibit at x_m=0.6 a change of slope which becomes steeper. The slope increase implies that heat transfer augmentation is larger at high values of average quality, i.e., in the annular flow. It may be inferred that at low values of G the enhancement is more effective in stratified flow than in annular flow while at high x_m the situation reverses. Data for R22 are characterized by a constant slope, at least up to x_m=0.65, and values comparable to those of R134a at low quality, while becoming larger as x_m increases. The enhancement factor, not shown, displays an increasing trend with x_m similar for both the refrigerants but ranging between 1.4 and 1.8 for R134a, and between 1.9 and 2.3 for R22.

Figure 7 shows data of the pressure drop per unit length $\Delta p_c/L$ plotted versus the mass flux G obtained for the same conditions reported in Figure 3 (x_{in}=0.7 and Δx=0.4). As seen, pressure gradients are considerably higher for R134a than for R22 and moreover the increase with respect to the smooth tube is more pronounced for R134a. This result is clearly displayed in Figure 8 where the penalty factor, defined as the ratio of the pressure drop in the microfin tube to that in the smooth tube, is plotted versus G; values for R134a vary between 2.1 and 1.4 and hence they result from 27% to 10% higher than those for R22 which range between 1.7 and 1.3. Finally, pressure gradient is quite sensitive to variations of quality but weakly dependent on quality change; in both the cases, the data obtained for the same conditions reported in Figures 4 and 5 exhibit an increasing trend with x_m and Δx, respectively, and values higher for R134a than for R22. For R134a the penalty factor, not shown, is a decreasing function of the average quality x_m too, ranging between 2.2 and 1.4.

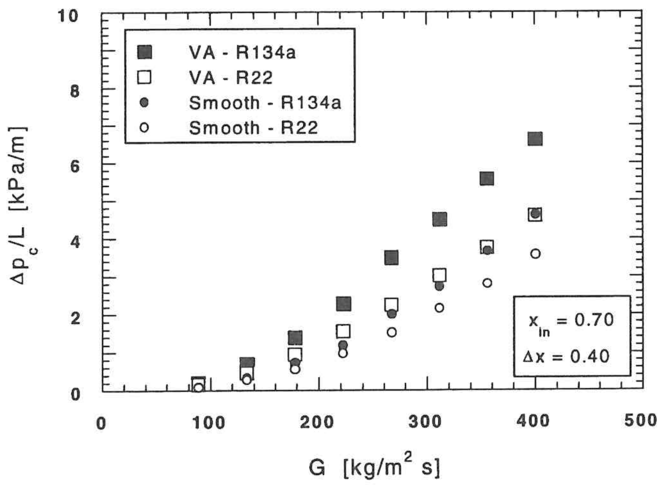

Figure 7. Pressure drop per unit length versus mass flux G for fixed inlet quality x_{in} and quality change Δx.

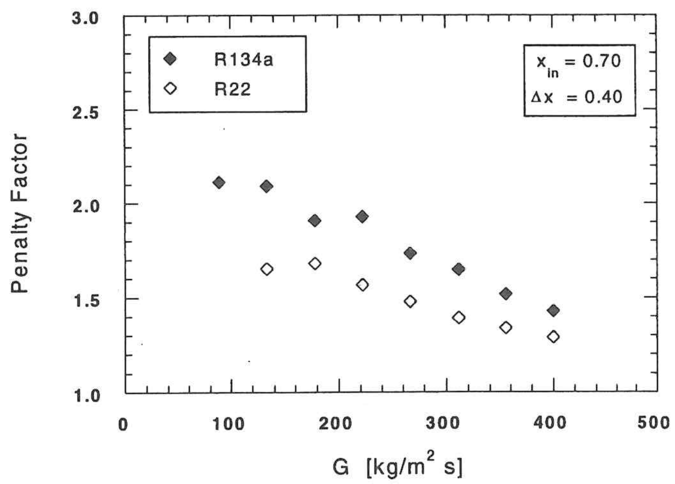

Figure 8. Pressure drop per unit length versus average quality x_m for fixed mass flux G and quality change Δx.

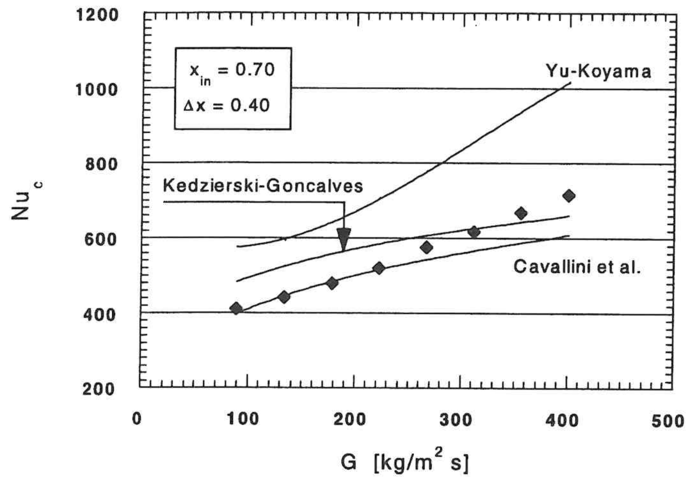

Figure 9. Experimental and calculated Nusselt number Nu_c versus mass flux G for fixed x_{in} and quality change Δx.

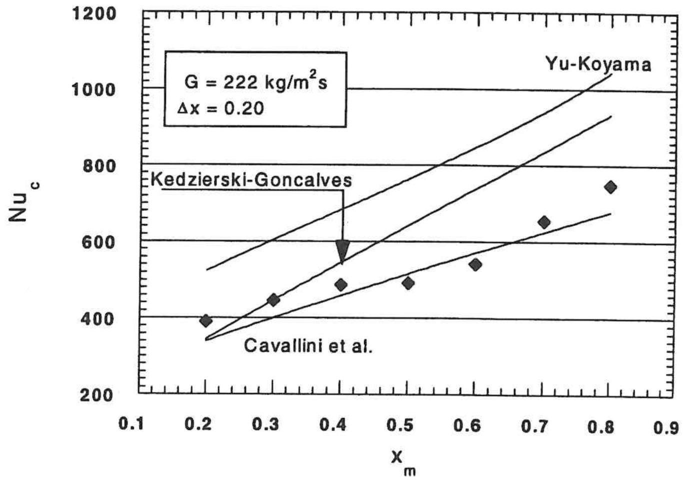

Figure 10. Experimental and calculated Nusselt number Nu_c versus average quality x_m for fixed mass flux and quality change.

COMPARISON WITH CORRELATIONS

Several correlations for condensation of refrigerants inside enhanced tubes were selected from the literature to carry out comparisons with the present data for R134a. For heat transfer, the correlations of Kedzierski and Goncalves [7], Yu and Koyama [8], and Cavallini et al. [10] are considered whereas the pressure drop data are compared to the predictions obtained by the correlations of Haraguchi et al. [5], Kedzierski and Goncalves [7], Cavallini et al. [10], and Nozu et al. [13]. Figure 9 compares the experimental and predicted Nusselt numbers plotted as function of the mass flux G, obtained at the same conditions of Figure 3 (x_{in}=0.7 and Δx=0.4). In Figure 10, the calculated Nusselt numbers are compared with the experimental data in a plot versus the average quality x_m; these values are for G=222 kg/m^{2}s and Δx=0.2, i.e., the same conditions of Figure 5. As can be seen, the predictions of the Cavallini correlation agree very well with the data and are more correct in trend than those obtained with correlations of Kedzierski and Goncalves, and of Yu and Koyama. With the Cavalliniet al. correlation, all of the data are predicted within ±20%, with a mean deviation E=3.4% and a standard deviation σ=6.5%. The most significant deviations occur at high mass flux, where the correlation tends to underpredict the experimental data. The mean deviation and the standard deviation of the Kedzierski and Goncalves correlation are 13.5 and 15 percent, respectively. This correlation tends to overpredict the experimental data, particularly at high average vapor quality and at low mass fluxes. The Yu and Koyama correlation is the worst predictor of the data with a mean deviation E=43.3% and the standard deviation σ=12.8%. This correlation consistently overpredicts all of the data with a deviation that exceeds 50% for some data.

Attention is now turned to the comparison of pressure drop results. Figures 11 and 12 compare the measured values of pressure drop with the predictions of the correlations as a function of G and x_m, respectively. The experimental conditions are the same as for Figures 9 and 10. It can be noticed, that none of the correlations satisfactorily predict the pressure drop. The Haraguchi correlation proves to be the best predictor with a mean deviation of –24.0 percent and a standard deviation of 11.4 percent, though it tends to underpredict the experimental data. The same tendency is exhibited also by the correlations of Cavallini et al. (E=–33.9% and σ=23.1%) and

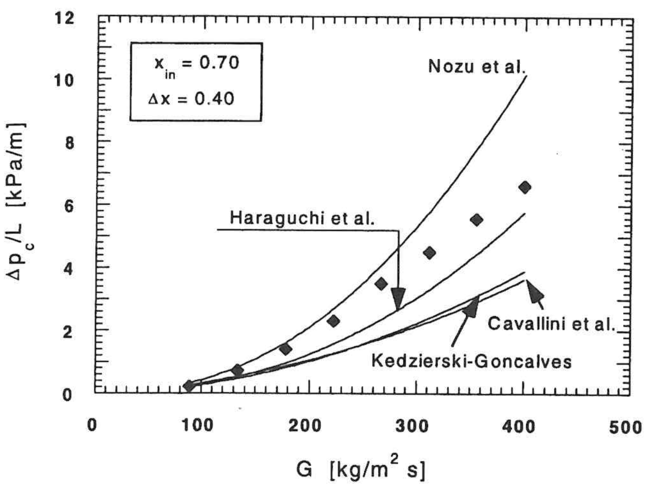

Figure 11. Experimental and calculated pressure drop per unit length versus mass flux G for fixed x_{in} and quality change Δx.

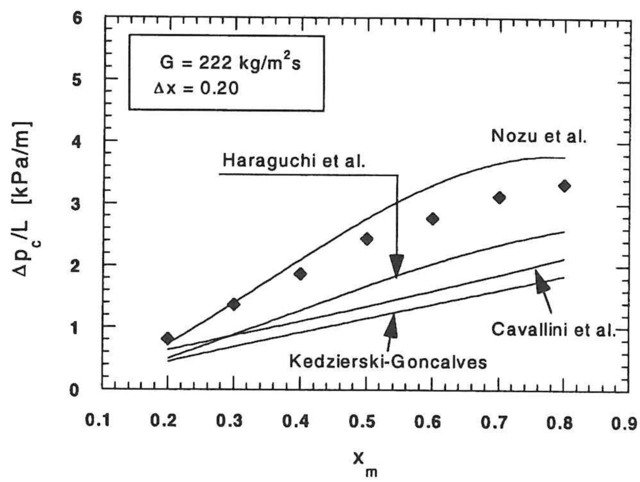

Figure 12. Experimental and calculated pressure drop per unit length versus average quality x_m for fixed G and Δx.

of Kedzierski and Goncalves (E=–40.5% and σ=15.8%). On the contrary, the Nozu correlation overpredicts the data because they exhibit a mean deviation of 26.0% with a σ=17.5%; despite this rather high deviation, though, the predictions of the correlation prove to be corrected in trend.

CONCLUSIONS

Heat transfer coefficient during convective condensation of the refrigerants R134a and R22 is quite sensitive to variations of mass flux G and average quality x_m, exhibiting with both of them an increasing trend, but it is a weakly dependent function of the quality change Δx. Results for R134a inside the microfin VA-tube are quite lower than those of R22. The enhancement factor as a function of G displays a maximum and varies between 1.5 and 1.25 for R134a, and from 2.1 to 1.8 for R22. As a function of x_m, instead, the enhancement factor exhibits an increasing trend with values within 1.4 and 1.8 for R134a, and within 1.9 and 2.3 for R22.

Regarding pressure drop, the refrigerant R134a shows values higher than those of R22 for both the smooth and microfin tubes, when comparison is made at the same values of G, x_m and Δx. What is more, the increase in pressure drop between the microfin tube and the smooth tube is more pronounced for the refrigerant R134a. The penalty factor as a function of G varies within the ranges of 2.1-1.4 and 1.7-1.3 for R134a and R22, respectively; consequently, the penalty factor for R134a is 10-27% higher than that for R22.

Data are compared to predictions of some correlations recently proposed for enhanced tubes. For heat transfer, the Yu and Koyama correlation consistently overpredicts the present data with a mean deviation of 43.3%; on the contrary, with the Cavallini et al. correlation, all of the data are predicted within ±20% with E=3.9% and σ=6.5%. Finally, the Kedzierski and Goncalves correlation tends to overpredict the data particularly at high x_m and at low G (E=13.5% and σ=15%). For pressure drop, none of the correlations here considered (Haraguchi, Kedzierski and Goncalves, Cavallini, and Nozu) satisfactorily predict our data. The Haraguchi correlation, though tends to underpredict data, proves to be the less inaccurate predictor with E=–24.0% and σ=11.4%.

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