Some Aspects of Experimental in-Tube Evaporation

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The heat transfer characteristics of refrigerant-oil mixture for horizontal in-tube evaporator have been investigated experimentally. A smooth copper tube and a micro-fin tube with nominal 9.5 mm outer diameter and 1500 mm length were tested. For the pure refrigerant flow, the dependence of the axial heat transfer coefficient on quality was weak in the smooth tube, but in the micro-fin tube, the coefficients were 3 to 10 times greater as quality increases. Oil addition to pure refrigerant in the smooth tube altered the flow pattern dramatically at low mass fluxes, with a resultant enhancement of the wetting area by vigorous foaming. The heat transfer coefficients of the mixture for low and medium qualities were increased at low mass fluxes. In the micro-fin tube, however, the addition of oil deteriorates the local heat transfer performance for most of the quality range, except for low quality. The micro-fin tube consequently loses its advantage of high heat transfer performance for an oil fraction of 5%. Results are presented as plots of local heat transfer coefficient versus quality.

Key Words : Refrigerant-Oil Mixture, In-Tube Evaporator, Smooth Tube, Micro-Fin Tube, Heat Transfer Coefficient, Mass Flux

Nomenclature

D_e : Envelope diameter (mm)
D_o : Outside diameter
f : Fin height (mm)
G : Mass flux (kg/(m²s))
q : Heat flux (kW/m²)
t : Tube thickness

1. Introduction

Throughout the past forty years, a great deal of heat transfer augmentation (enhancement) technology has been developed to improve heat transfer performance. Micro-fin tubes, which have the inside surface enhanced with numerous fins (more than 50 internal fins for 9.5 mm outer diameter) and very small fin height (less than 2.5% of the inside diameter) have recently become popular in evaporators and condensers for the refrigeration and air-conditioning industry. Several papers from Japan reported the evaporation heat transfer enhancement, varying from 1.5 to 10 depending on micro-fin tube geometry and system conditions (Ito et al. 1977, Tatsumi et al. 1982). Detailed investigations of the evaporation heat transfer in a micro-fin tube with both pure refrigerant and refrigerant-oil mixtures were reported by Schlager et al. (1988, 1989). Heat transfer coefficients in the micro-fin tubes, based on a nominal equivalent smooth tube area, were 1.6 to 2.2 times greater than those in a smooth tube. Although micro-fin tubes have been effectively used in real applications, their local heat transfer characteristics have not yet been clearly delineated. Another consideration is that oil is present in refrigeration systems to lubricate compressor moving parts. In the smooth tube, the heat transfer coefficient at the low mass flux increased as the oil mass fraction increased up to 2.5~5% (Kim et al. 1988). However, for a high mass flux, the heat transfer coefficients decreased at an oil fraction of 5% (Schlager et al. 1989, Eckels and Pate 1991).

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Recently, interest in understanding the local heat transfer mechanisms in evaporators has been increasing because of the need to replace conventional refrigerants due to the perceived ozone depletion problem. Torikoshi et al. (1992) obtained the averaged evaporation heat transfer coefficients for a R-134 a/PAG oil mixture where R-134a is the most promising alternative to R-12. Their results indicated that the evaporation heat transfer coefficients for R-134a without oil were about 25% higher than those for R-12 at the same mass fluxes, while the dependence on mass flux was similar to that for R-12. For a micro-fin tube, the heat transfer coefficient with oil increased at small oil fraction but decreased at large oil fraction. Torikoshi and Ebisu (1993) extended the experimental measurement of the averaged heat transfer coefficient for R-32, R-134a, R-22, and a mixture of R-32/134a. The observed increase or decrease in the evaporation heat transfer coefficient for alternative refrigerant compared to R-22 was ascribed to the difference in refrigerant properties. However, Ha and Bergles (1993a) reported that for an oil fraction of 5% and a mass flux of $50 kg/(m^2s)$, the heat transfer performance with the micro-fin tube is below that for the smooth tube with oil. Kim et al. (1997) analyzed the dynamic characteristic of a smooth tube evaporator with respect to refrigerant flow rate, inlet enthalpy, inlet air velocity and air temperature. The model for the dynamic characteristics in two phase region may be used to determine optimum heat exchanger design parameters if sufficient data for micro-fin tubes can be provided. Because most of the previous work includes different system conditions, heating methods, and tube geometries, with rare flow pattern analyses, there have been conflicting results for the heat transfer coefficient and its mechanical interpretation, even in the case of pure fluids. Acknowledging that any refrigerant would behave the same way, the reliable results for R-12 would be qualitatively transferable to alternative refrigerants. This paper is mainly directed toward understanding the local evaporation heat transfer mechanism of refrigerant-oil mixtures.

2. Problem Simulation

In order to determine the detailed heat transfer and fluid flow characteristics of actual heat exchangers in refrigeration and air-conditioning systems, laboratory studies simulate actual in-tube evaporators using electric heating, air heating with fins, or liquid heating. Advantages and disadvantages of various simulation methods are summarized in Table 1.

The thermal boundary condition on the outside tube of the typical air-cooled evaporator coil was simulated, wherein refrigerant flows inside the tubes and the heating air flows across the tubes. If the air-side convective heat transfer coefficient is nearly uniform, the heat flux supplied by the air can be assumed uniform. This is because the axial wall temperature variation is negligible compared to the average temperature difference between the surrounding air and the wall. The fact that there is a temperature gradient in the fin does not affect this supposition. The uniform heat flux (UHF) condition is apparently applicable, except in the dry-out region. Thus, the indirect electric heating method can be used unless practical air-cooled evaporators are tested in a wind tunnel. The wire-wrap technique was chosen, and this has an additional advantage in that the somewhat discontinuous axial heating provided by the fins can be simulated. Meanwhile, the local heat transfer coefficient of refrigerant flow is defined in terms of the inside heat flux and the temperature difference between the inside wall and the fluid. It should be noted that if the tube interior is fully wetted, such as in the annular flow pattern, the evaporator capacity is high, because of the more uniform circumferential heat transfer coefficient. Partially wetted flow patterns have circumferentially nonuniform inside heat flux because of the different order-of-magnitude of heat transfer coefficients in the vapor and liquid phases. The magnitude of this effect depends on the material and geometry of the test tube.

For actual evaporator copper tubes, it is expected that axial heat flux can be assumed uniform for proper wire spacing, while the circumferential

Table 1 Comparison of heating methods for horizontal in-tube evaporation

Method	Advantage	Disadvantage
Electrical resistance wire heating	-Uniform heat flux -Axially local heat transfer coefficient -Easy to control power	-Wall temperature excursion at dryout -Limited space for thermocouple attachment -Small-scale variation in axial heat flux -Heating wire interaction with thermocouple (local hot spot)
Direct electric heating	-Uniform heat flux -Easy to analyze inside heat flux and axial heat transfer coefficient -Flow observation possible using glass tube with resistance coating inside or outside of tubes	-Uncertainties of wall temperature measurement with electrical insulation -Axial heat conduction from bus bars -Limited tube selection (need high current for copper tubes) -Temperature excursion at dryout -Possible non-uniform heat flux for resistance-coated glass tube -Inside coating could alter surface characteristics for boiling
Liquid heating	-Nearly constant wall temperature -LMTD average heat transfer, sectional average, or local heat transfer -Close to actual application with heat flux	-Difficulty to determine local heat transfer -Hard to control variables (mass flux and heat flux are depependent) -Wilsion plot may be needed to obtained the average heat transfer coefficient of the heating fluid
Air heating with fins	-Best simulation for actual air-cooled evaporator	-Virtually impossible to obtain local heat transfer -Need cross-flow arrangement (wind tunnel to provide air) -Thermocouple attachment problem

heat conduction will be substantial. A relatively large length, if the electrical heating method is adapted, should also be used to simulate the moderate heat flux in an actual evaporator. An analysis of the axial and circumferential heat flux variation as a function of the tube characteristics and flow conditions was performed by Ha and Bergles (1993b).

3. Experimental Program

The experimental rig was designed and constructed for a series of heat transfer measurements. Refrigerant-12 (R-12) was selected as a working fluid because R-12 data are important as reference data for alternative refrigerants, such as R-134a. Due to its popularity and miscible compatibility with R-12, a naphthenic base mineral oil, SUNISO 3GS (150SUS viscosity), was chosen as the test oil in the refrigerant-oil mixture tests. The experimental apparatus consisted of two main subloops: a refrigerant flow control loop and an oil flow control loop. Details of the complete test apparatus can be found in Ha and Bergles (1993a).

The horizontally mounted test evaporator was a straight copper tube (nominal 3/8 inch air-conditioning tubing) with outer diameter of 9.5 mm and total length of 1.5 m. The exit end of the test section had a sight glass of the same inside diameter as the test section to observe the flow pattern. Figure 1 shows geometrical parameters of a micro-fin tube. Table 2 lists dimensions of the smooth tube and the micro-fin tube tested in this

Table 2 Geometrical parameters of the smooth and the micro-fin tube

Parameters	Smooth tube	Micro-fin Tube
Outside diameter (mm), D_o	9.5	9.5
Envelope diameter (mm), D_e	8.0	8.94
Tube thickness (mm), t	0.75	0.29
Fin height (mm), f		0.18
Spiral angle (degrees)		18
Number of fins		60

Table 3 Operating parameter ranges for in-tube evaporation

Parameter	Value
Mass flux, kg/(m² s)	50, 75, 100, 176
Heat fluxes, kW/m²	5, 7. 5, 15, 20
Inlet quality	0.1−0.6
Exit quality	0.4-Superheated
Evaporating pressure, MPa	0.318-0.321
Oil mass fraction, %	0, 1, 3, 5

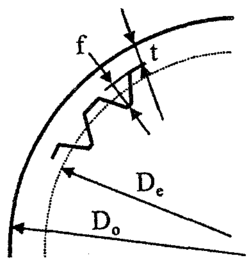

Fig. 1 Geometrical parameters of a micro-fin tube

study. Mean diameter, which is used to calculate mass flux and inside heat flux for a micro-fin tube, is defined as the diameter of the corresponding smooth tube that has the same cross-sectional area. The wall temperatures were measured by means of 36 gage (0.127 mm diameter) T-type thermocouples at ten axial locations, where the axial direction refers to the direction of flow. At each axial location, five thermocouples were placed on half of the tube 45 degrees apart, circumferentially, except at two axial locations, where eight thermocouples measure the entire circumferential wall temperatures to verify the symmetry of the temperature distribution. Heat to the test section was indirectly supplied by means of electrical resistance elements (20 AWG nichrome wire with fiber glass braid insulation). The tube was then heavily insulated with three layers of 25 mm thick fiber glass to minimize heat gain or loss to the tube. In these experiments, "steady-state" conditions were defined by less than 1% fluctuations of system pressure, mass flux and heat flux. It usually took about one-half hour to establish steady-state conditions after start-up. Each test run consisted of a set of three readings of all instruments approximately one-half hour after equilibrium conditions had been obtained. Internal flow area and inside heat transfer surface based on envelope diameter, D_e in Fig. 1, was used to calculate mass flux, G, and inside heat flux, q_i.

For smaller systems, such as window air conditioners, the cooling capacities and refrigerant mass fluxes, normally 100~250kg/(m²s) for 1~5 kW are interrelated. However, inverter-controlled air-conditioners are becoming more popular due to their ability to adjust the cooling load corresponding to indoor and outdoor climates. They have a wide mass flux range that extends as low as 25kg/(m²s) during operation. A summary of the experimental operation conditions is presented in Table 3.

The amount of oil in the refrigerant downstream of the separator was measured using the boiling-off method suggested by ASHRAE Standard (1984). When the oil mass fraction was found to be less than 0.1%, the refrigerant was referred to as oil-free. For refrigerant-oil mixture tests, in order to verify the oil mixture concentration, a sample of the refrigerant-oil mixture was taken after the mixing section for selected tests. The measured oil mass fraction after boiling off the refrigerant was accurate to within ±5% of the nominal value.

The vapor pressure of the oil is extremely low so that the oil remains in the liquid mixture as evaporation processes. This leads to an elevation of the saturation pressure of the mixture. Therefore, there are two kinds of the local heat transfer coefficient: one using the pure refrigerant saturation temperature as the bulk temperature, and the other using the mixture saturation temperature as the bulk temperature. Note that 1℃ deviation may be important in calculating the heat transfer coefficient because the wall superheat is small in practical application. In this study, the pure refrigerant saturation temperature was used in calculating the heat transfer coefficient, for simplicity. Details of saturation temperature of refrigerant-oil mixture can be found in Ha (1992).

Evaporating pressures at the entrance and the exit of the test section were measured with strain-gage-type absolute pressure transducer. These transducers were calibrated for the operating range by a dead weight tester and showed an accuracy of 0.25% near the operating pressure of 0.32 MPa. The saturation temperature of pure refrigerant based on pressure was compared with bulk temperature from sheathed thermocouples inserted into the flow stream and showed a maximum deviation of 0.3℃. Linear extrapolation of the saturation temperature associated with pressure drop along the test section was used for local fluid temperature. It is noted that the averaged evaporating pressure for various test condition kept constant and pressure drop was not so large for small quality change in the test section.

In this study, an energy balance for pure refrigerant flow on the evaporator was computed with the subcooled inlet at the subcooler and the superheated condition at the exit of the test section. An overall error in the energy balance of $\pm 5\%$ was accepted during the test. The pure vapor quality of fluid in the test section was then calculated by energy balance. The local heat transfer coefficients were obtained simply by neglecting the circumferential heat conduction effects, because the wall temperatures were rather uniform circumferentially. This was calculated by dividing the uniform heat flux by the circumferentially averaged wall superheat.

The uncertainty of the mass flux and the heat flux are $\pm 1\%$ and $\pm 2\%$, respectively. For the local quality, the uncertainty is a function of the axial length, ranging from $\pm 0.5\%$ to 2%. The initial oil fraction has an uncertainty of $\pm 5\%$. Based on sample calculations for the smooth tube with refrigerant-oil mixtures, the uncertainty of the heat transfer coefficient was estimated as slightly less than $\pm 4\%$.

4. Experimental Results

4.1 Smooth tube with pure refrigerant

4.1.1 Mass flux effects

Mass flux effects at low heat flux are shown in Fig. 2. The heat transfer coefficient increases as mass fluxes increase from 50 kg/(m^{2}s) to 100 kg/(m^{2}s) over the entire quality range. Doubling mass flux increases the heat transfer coefficient by 20-50%, depending on the quality. The data apparently gives a nearly constant heat transfer coefficient at low mass flux; this is probably due to the dominance of nucleate boiling. It is noted that the correlation by Kandlikar (1990) overpredicts the mass flux effect because the predicted coefficients are different from the measured data except case (1) in Fig. 2. Compared with the results from the present experiment data, convection term in the Kandlikar correlation is too much increased, and the disparity becomes larger as mass fluxes increase.

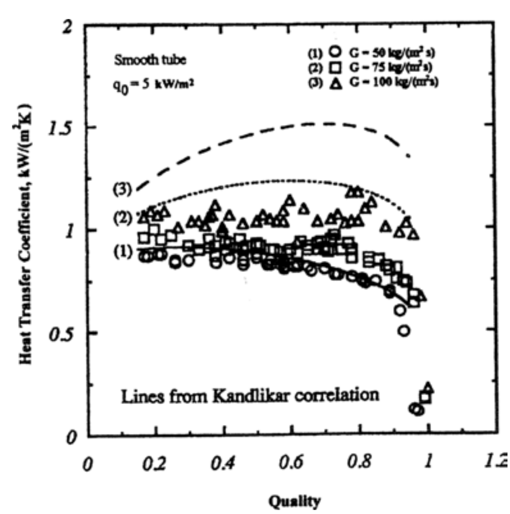

Fig. 2 Mass flux and quality effects on the heat transfer coefficient for smooth tube with pure refrigerant at a low heat flux

4.1.2 Heat flux effects

For a constant mass flux of 50 kg/(m^2s), the heat transfer coefficient is quite independent of the local quality up to 0.8 for various heat fluxes in Fig. 3. This figure indicates that increasing heat flux increases the heat transfer coefficient to a large extent, with almost parallel shifting of the curves up to a quality of 0.8. The heat transfer coefficients are increased by 70% for heat fluxes from 5 kW/m^2 to 10 kW/m^2 and 20% for heat fluxes from 7.5 kW/m^2 to 10 kW/m^2. It is expected that the dependence of the heat transfer coefficient on heat flux is suppressed as mass flux increases, and there may be effective heat flux for a given mass flux. Meanwhile, it seems that the Kandlikar correlation suggests a lesser contribution of the heat flux effect on the coefficients, and the predicted coefficients approach single values regardless of heat flux variation, which is substantially different from the measured data. Thus, the disparity of the coefficients from the correlation and the experiment becomes greater as quality increases for high heat fluxes. It seems that for a partially wetted flow, increasing heat flux can play a role not only to promote the nucleate boiling but also to enhance the liquid wetting characteristics, possibly due to a change in surface tension at the wall.

4.2 Smooth tube with refrigerant-oil mixtures

Flow patterns of refrigerant-oil mixtures in smooth tubes show that an increase in oil fraction promotes foaming, leading to increased liquid wetting at the wall. It was observed that at a quality of 0.8, a high vapor velocity leads to a strong agitation of the wavy foam, which results in frequent transportation of the foam slug from the main liquid body toward the sides and the top of the tube wall and, finally, establishes complete wetting around the entire perimeter. A liquid film formed on the tube wall is not homogeneous, the upper part being oil-rich and highly viscous while the oil content of the lower part apparently corresponds to that of the bulk of liquid.

The local transfer coefficients, for a mass flux of 50 kg/(m^2s) and a heat flux of 5 kW/m^2, are presented in Fig. 4. Showing scattered values possibly attributed to the less steady flow behavior of refrigerant-oil mixtures. Heat transfer coefficients for oil fractions of 3% and 5% are apparently similar. Both gradually decrease above qualities of 0.5, and are finally less than that for the pure refrigerant for qualities above 0.8. Data for 1% oil addition indicate that the heat transfer enhancements extend to a quality higher than 0.9.

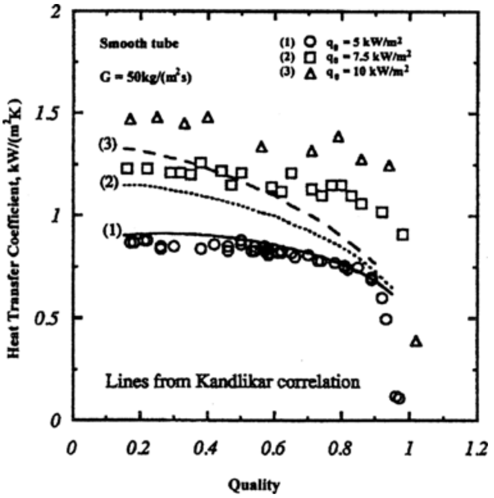

Fig. 3 Heat flux effect on the heat transfer coefficient for smooth tube with pure refrigerant at low mass flux

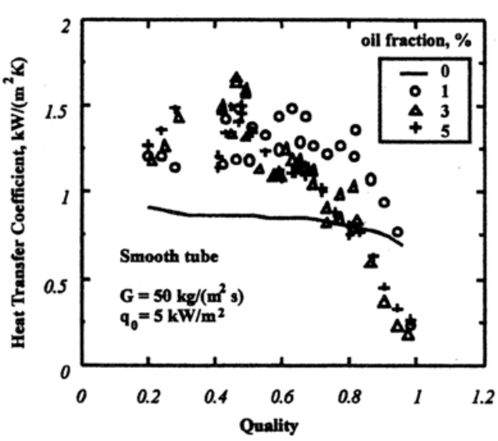

Fig. 4 The local heat transfer coefficients of refrigerant-oil mixtures for 5% oil fraction: smooth tube at a low mass flux of 50 kg/(m^2 s) and a heat flux of 5 kW/m^2

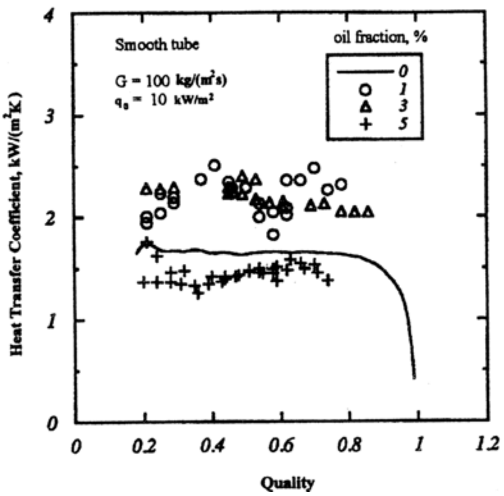

Fig. 5 The local heat transfer coefficients of refrigerant-oil mixtures for the smooth tube at a moderate mass flux of 100 kg/(m^2 s) and a heat flux of 10 kW/m^2

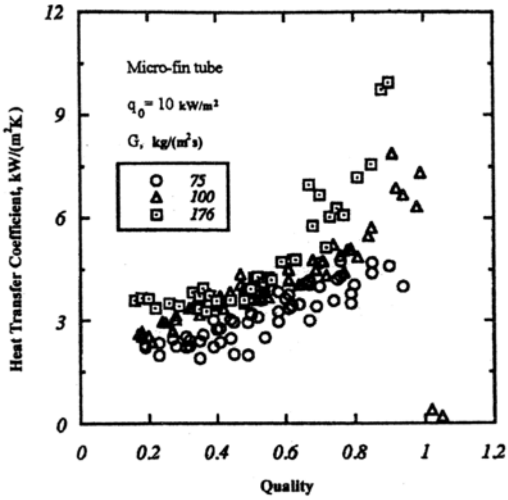

Fig. 6 Mass flux effect on the axial heat transfer coefficient of the micro-fin tube with pure refrigerant flow at a heat flux of 10 kW/m^2

Based on the data in this figure, the local heat transfer coefficients are substantially enhanced by about 1.5 times up to a quality of 0.6.

The local transfer coefficients are is shown in Fig. 5 for a mass flux of 100 kg/(m^{2}s) and a heat flux of 10 kW/m^2. As compared with previous data at low mass fluxes, the heat transfer coefficient enhancement for 1% and 3% oil fractions are about 1.3, but data for 5 % oil fraction indicate that the heat transfer deteriorates even at low qualities.

4.3 Micro-fin tube with pure refrigerant

4.3.1 Mass flux effects

The mass flux effect on the local heat transfer coefficient for a low heat flux of 10 kW/m^2, is shown in Fig. 6. This figure indicates that a mass flux increase from 100 kg/(m^{2}s) to 176 kg/(m^{2}s) results in a relatively small change in heat transfer coefficient for a quality lower than 0.6, except for a very low quality, but there is 1.5 times increase for a quality above 0.6. The peak heat transfer coefficient is encountered at a quality close to 1. 0, and some scattering seems to be attributed to the liquid film fluctuation. In any case, the local heat transfer coefficient is increased in an apparently linear fashion with increasing quality up to 0.8, which is not obtained in the smooth tube for low mass fluxes. It is thought that in order to obtain the maximum heat transfer performance, first, an uniform liquid film distribution along the tube perimeter is required, which is dominantly controlled by mass flux and quality. Secondly, thinning of the liquid film in the fine grooves is advantageous.

4.3.2 Heat flux effects

The dependence of local heat transfer coefficients on the heat flux for a mass flux of 100 kg/(m^2 s) is shown in Fig. 7. An increase in heat flux from 5 kW/m^2 to 20 kW/m^2 leads to a minor enhancement of the heat transfer coefficient at a quality lower than 0.8. However, for qualities higher than 0.8, an increase in heat flux from 10 kW/m^2 to 20 kW/m^2 leads to a reduction in the peak heat transfer coefficient and a shift in that peak to the upstream direction. Heat flux should play a negligible role in the local heat transfer, unless the boiling mechanism is dominant. However, at high qualities, dryout is evident, and the dryout process is obviously affected by the level of heat flux, as in conventional evaporator tubes.

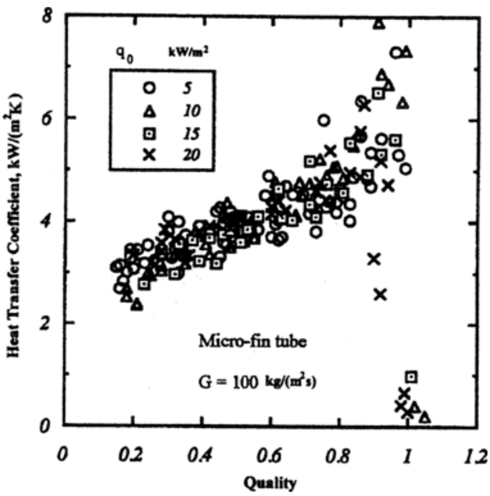

Fig. 7 Heat flux effect on the axial heat transfer coefficient of the micro-fin tube with pure refrigerant flow at a mass flux of 100 kg/(m² s)

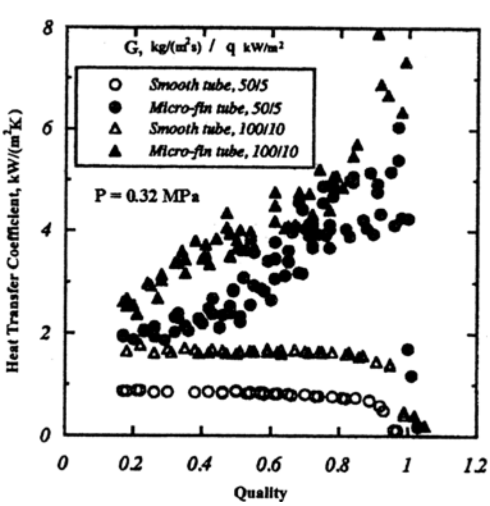

Fig. 8 Comparison of the heat transfer coefficient of the smooth tube and the micro-fin tube with pure refrigerant flow for operating conditions of 50 kg/(m² s) with 5 kW/m², and 100 kg/(m² s) with 10 kW/m²

4.3.3 Enhanced tube effect

The comparison of the local heat transfer coefficient of the micro-fin tube with the smooth tube data is shown in Fig. 8. This figure clearly shows that the heat transfer coefficient is significantly enhanced in the micro-fin tube for similar mass and heat flux conditions. The enhancement

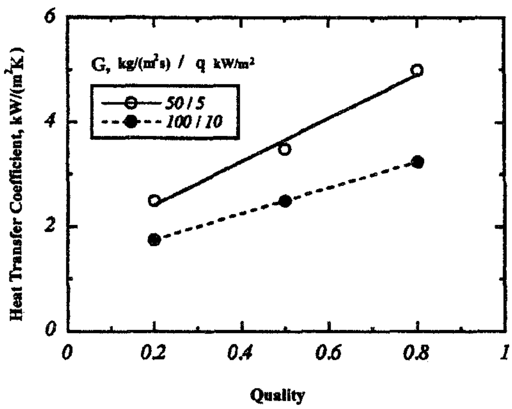

Fig. 9 Heat transfer coefficient enhancement of micro-fin tube with a reference of data for the smooth tube with pure refrigerant flow for 50 kg/(m² s), 5 kW/m² and 100 kg/(m² s), 10 kW/m²

becomes higher for high qualities, because the heat transfer coefficient in the micro-fin tube is increased with an increase in the quality, while smooth tubes have a rather constant heat transfer coefficient throughout most of the quality range. However, the heat transfer enhancement of the micro-fin tube decreases with increasing mass flux, as indicated in Fig. 9. Yoshida et al. (1988) also reported such a trend that with increasing mass flux above 100 kg/(m²s), the enhancement is rapidly reduced, and the heat transfer coefficients are close to the values for a smooth tube for 300 kg/(m²s).

4.4 Micro-fin tube with refrigerant-oil mixture

Experimental data for the local heat transfer coefficient for refrigerant-oil mixtures in a micro-fin tube were previously reported by the present authors. The following data are mainly from Ha and Bergles (1993a) for a systematic comparison of oil effect on smooth tube and micro-fin tube. For a doubling of the mass flux from 50 kg/(m²s) to 100 kg/(m² s), the enhancement ratio is increased to 1.8 for an oil fraction of 3% and 2.0 for an oil fraction of 5% over the quality range of 0. 2 to 0.8, assuming no heat flux effect. Note that heat transfer surface for the micro-fin tube is 50% larger, due to numerous internal fins, than that of

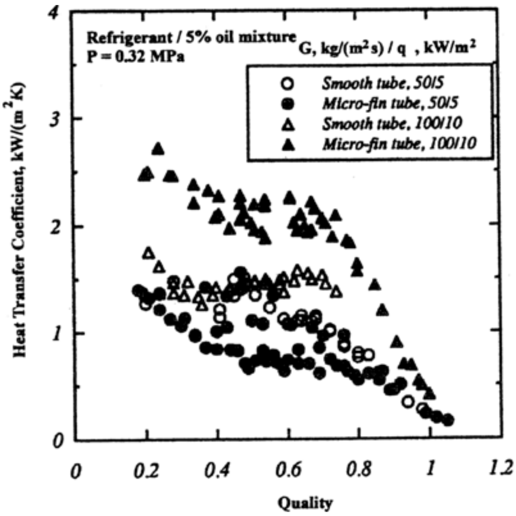

Fig. 10 Comparison of the heat transfer coefficients of the smooth tube and the micro-fin tube with refrigerant-oil mixtures flow for operating conditions of 50 kg/(m² s) with 5 kW/m², and 100 kg/(m² s) with 10 kW/m² (Ha and Bergles, 1993a)

the smooth tube. For an oil fraction of 5% shown in Fig. 10, the heat transfer coefficient of the micro-fin tube is lower than that of the smooth tube for a low mass flux of 50 kg/(m² s), but it is increased by a factor of 1.5 for a moderate mass flux of 100 kg/(m² s).

From the flow visualization, a heavy and viscous oil-rich film around the entire perimeter, with suppressed foaming activity, was observed. This film is an indication of the oil concentration built up at the phase interface. The micro-fin grooves might dampen the wavy characteristics so that foaming activity is suppressed. Even though some scatter in the heat transfer coefficient should be considered, the presence of oil in micro-fin tube significantly deteriorates the heat transfer performance, and it finally leads to the disadvantage of the micro-fin tube for low mass fluxes.

5. Conclusions

(1) In smooth tubes with pure refrigerant at low mass fluxes, the axial heat transfer coefficient increased with increasing heat flux and mass flux. Its dependence on quality was weak for smooth tubes with pure refrigerant up to the dry-out region for low mass fluxes, and it tended to increase with increasing quality for high mass fluxes.

(2) Oil addition to pure refrigerant in smooth tubes altered the flow pattern dramatically at low mass fluxes with enhancement of the wetting area. The heat transfer coefficients of the mixture for low and medium qualities were 1.2 to 1.5 times higher than that of the corresponding pure refrigerant cases at low mass fluxes, depending on the oil fraction, but the degree of enhancement was decreased with an increase in mass flux. However, the enhancement factor went down below 1.0 for the high quality region, especially for moderate mass fluxes, due to the insulating layer resulting from the oil concentration gradient at the phase interface.

(3) Micro-fin tubes with pure refrigerant flow at low mass fluxes enhanced the liquid wetting area due to the spiral grooves. The heat transfer coefficients were remarkably increased with increases in quality due to thinner liquid film thickness. The heat transfer was enhanced by factors of 3 to 10, depending on the quality, for low mass fluxes, but the ratio was reduced with an increase in mass flux.

(4) The addition of the oil in refrigerant flow for the micro-fin tube deteriorated the heat transfer coefficient, with the rapid decrease at high qualities, and the negative effect tend to be less severe at higher mass fluxes. Heat transfer coefficients were quite constant for 1% oil addition, and gradually decreased for 3% an 5% oil addition with increasing quality.

(5) The heat transfer coefficient in the smooth tube at low mass fluxes was strongly dependent on the heat flux. With increasing mass flux, there was less effect of heat flux as the liquid phase convection became the superior mechanism. However, the heat transfer coefficient was a weak function of heat flux in the micro-fin tube, except for very high qualities where dryout occurred.

(6) The heat transfer coefficient was a moderate function of mass fluxes due to the enhancement of both the liquid phase convection heat transfer and liquid wetting area for the smooth

and the micro-fin tube. However, the enhancement became less once the fully wetted liquid film distribution was established.

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