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Condensation heat transfer for R-22 and R-407C refrigerant–oil mixtures in a microfin tube with a U-bend

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Abstract

Condensation heat transfer experiments for two refrigerants, R-407C and R-22, both mixed with polyol-ester (POE) and mineral oils were performed in straight and U-bend sections of a microfin tube. Experimental parameters included oil concentration (varied from 0% to 5%), mass flux (varied from 100 to 400 kg/m^2 s) and inlet quality (varied from 0.5 to 0.9). The enhancement factors (EFs) for both R-22 and R-407C refrigerants at the first straight section decreased continuously as the oil concentration increased. They decreased rapidly as mass flux decreased and inlet quality increased. The heat transfer coefficients had the maximum at the 90° position of the U-bend. The heat transfer coefficients in the straight section after the U-bend, within a length 48 times that of the inner tube diameter, were larger by a maximum of 33% than the average heat transfer coefficient in the straight section before the U-bend.

Keywords: Condensation heat transfer; R-407C; R-22; POE oil; U-bend; Microfin tube

1. Introduction

R-407C (a mixture of R-32/R-125/R-134a, 23%/25%/52% by weight) is a short-term replacement of R-22 because its properties are similar to those of R-22. Refrigeration oil is often used for lubricating and sealing the compressor in refrigeration systems. It circulates with the refrigerant because there is no oil separator in a small refrigeration system. Mineral oil has been used for non-polar R-22 refrigerant, whereas polyol-ester (POE) oil has been used for polar refrigerant mixtures such as R-407C due to its solubility and miscibility. A microfin tube has been widely used as heat exchangers. Schlager et al. [1] reported that the microfin tube enhanced heat transfer by 50–100% compared with a smooth tube.

The effect of refrigeration oils on condensation heat transfer has been investigated by a number of researchers, who reported the degradation of the condensation heat transfer coefficient due to the presence of the refrigeration oil. Schlager et al. [2,3] and Eckels et al. [4,5] investigated the effect of refrigeration oil on condensation heat transfer in a microfin tube, while Tichy et al. [6] and Shao and Granryd [7] reported the effect of oil on condensation heat transfer in a smooth tube. Schlager et al. [3] found that, for 150 and 300 SUS oils, condensation heat transfer for R-22 decreased by 30% at 5% oil concentration. Eckels et al. [5] reported that the condensation heat transfer coefficients for R-134a degraded by 18% at 5% concentration of 150 SUS ester oil.

Heat exchangers for air-conditioning systems are often constructed of tubes containing U-bends. Heat-transfer characteristics in the U-bend are quite different from those in the upstream straight section, and they may affect heat-transfer characteristics in the downstream straight section. However, condensation heat transfer coefficients for a refrigerant–oil mixture has been reported only in straight smooth and microfin tubes. To the authors' knowledge, the effect of refrigeration oils on condensation heat transfer performance in a microfin tube containing a U-bend has not been reported in the literature.

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Nomenclature

c	oil concentration (%)
EF	enhancement factor
F	y-coordinate in Taitel and Dukler's flow pattern map
f	Darcy friction factor
G	mass flux (kg/m^2 s)
h	heat transfer coefficient (kW/m^2 K)
h^+	ratio of heat transfer coefficient at the second straight section to average heat transfer coefficients at the first straight section
L^+	ratio of length to inner diameter of the tube
q	heat flux in the test section (kW/m^2)
T	temperature (°C)
X_{tt}	Martinelli parameter
x	quality
y	mass fraction

Greek symbols

δ	uncertainty
ρ	density (kg/m^3)
μ	viscosity (kg/m s)

Subscripts

f	liquid
g	vapor
in	inlet of test section
lo	local
m	refrigerant–oil mixture
o	oil
r	refrigerant
wi	inner wall of test tube

The objective of the present study was to experimentally investigate the effects of both POE oil (315 SUS) and mineral oil (290 SUS) on the condensation heat transfer performance of R-407C and R-22 refrigerants in both straight and U-bend sections of a microfin tube.

2. Experimental apparatus and procedure

Fig. 1 shows a schematic diagram of the present experimental system, which consists of a refrigerant flow loop and two coolant flow loops. The refrigerant flow loop contains a test section, a pre-heater, a plate heat exchanger, oil injection and sampling devices, a magnetic refrigerant pump and a Coriolis mass flow meter. The coolant flow loops contain a constant-temperature bath, a chiller and rotameters. Fig. 2 shows the details of the test section. The test section consists of a microfin tube with two straight sections (each 1 m long) and a U-bend (0.0390 m long) positioned vertically for downward flow and an acrylic plate grooved concentrically around the microfin tube. The microfin tube has an outer diameter of 9.52 mm, an inner diameter of 8.53 mm, a fin height of 0.2 mm, a fin number of 60, and a fin spiral angle of 18°. The inner surface area ratio of the microfin tube to a smooth tube with the same inner diameter is 1.51. The diameter of the curvature for the U-bend is 2.61 times the outer tube diameter.

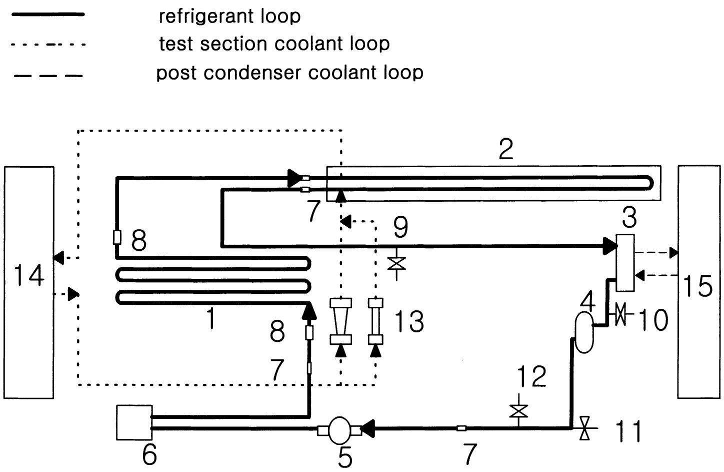

Fig. 1. Schematic diagram of the present experimental system [1 – pre-heater; 2 – test section; 3 – plate heat exchanger; 4 – receiver; 5 – refrigerant pump; 6 – mass flow meter; 7 – sight glass; 8 – Teflon tubing; 9 – refrigerant–oil mixture sampling valve; 10 – refrigerant inlet valve; 11 – drain; 12 – oil inlet valve; 13 – rotameter; 14 – constant temperature bath (120 l, 2 kW); 15 – chiller (3 kW)].

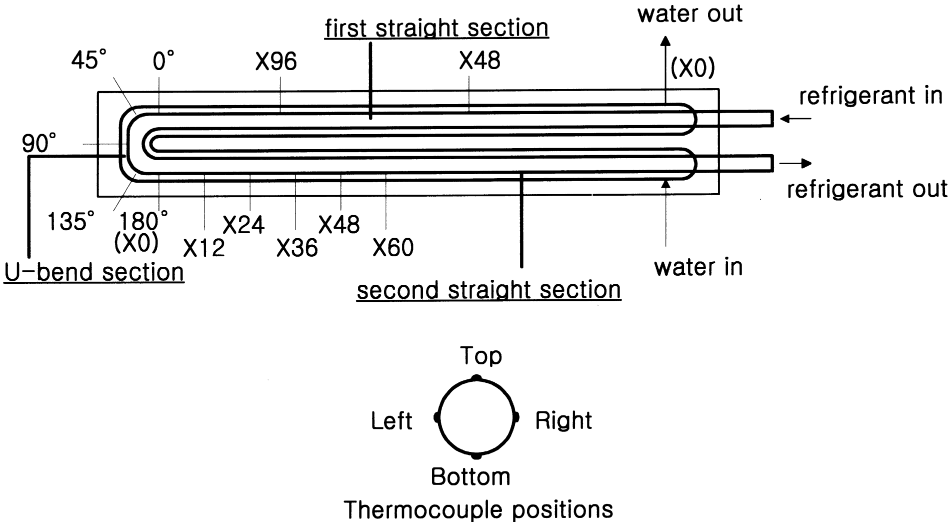

Fig. 2. Details of the test section.

The outer wall temperature was measured using T-type thermocouples (±0.15°C accuracy) at 12 positions (7 for the straight and 5 for the U-bend) along the direction of refrigerant flow. The thermocouple positions are shown in Fig. 2 by $\times\#$, where the number following $\times$ indicates the ratio of tube length from the inlet of the first (or second) straight section to the inner tube diameter for the straight section. The temperature was measured every 45° in the U-bent section of the tube. The outer wall temperatures were measured at four circumferential points (top, bottom, left and right) for each temperature measuring position and averaged for calculating the heat transfer coefficients. The inner wall temperature of the test section was estimated from the measured outer wall temperature by applying the radial heat conduction equation for a hollow cylinder.

The coolant temperatures were measured using T-type thermocouples (±0.05°C accuracy) at the inlet and outlet of the test section and those of the U-bend. The temperature rise of the coolant between the inlet and outlet of the test section was approximately 1°C. The heat flux on the test section ranged from 3.48 to 13.9 kW/m^2, and its uncertainty was within 5%, mainly due to the resolution of the thermocouples. The inlet pressure at the test section and pressure drop between the inlet and outlet of the test section were measured by both a pressure gauge (35 bar range, ±0.1% resolution) and a differential pressure gauge (350 mbar range, ±0.1% resolution), respectively.

Refrigeration oil was injected into the system through an oil injection port by using high-pressure nitrogen gas. The concentration of the oil was monitored by sampling the oil by using the boiling-off method suggested by the ASHRAE standard [8]. In order to measure oil concentration in the test system, the masses of the oil injector and sampling device before and after the oil injection and boiling-off processes were measured with a balance (4 kg range, 0.1 g resolution).

Table 1 shows the measured concentrations of the injected and sampled refrigeration oil. Since the difference between injected and sampled oil concentrations was within 0.1%, as shown in Table 1, the oil was assumed to be well-mixed and uniformly distributed in the refrigerant flow loop. The sampled oil concentrations were used for data analysis.

Since the vapor pressure of the oil was negligibly small compared to that of the refrigerant, the oil was

Table 1
Measured concentrations of injected and sampled refrigeration oil (weight%)

Oil concentration	R-22		R-407C	
	Injected (%)	Sampled (%)	Injected (%)	Sampled (%)
1	1.08	1.02	1.15	1.12
3	3.03	2.93	3.10	3.07
5	5.03	4.97	4.98	4.91

Table 2
Local oil concentrations calculated from Eq. (1) for the liquid phase of the refrigerant–oil mixture (by weight%; unit: %)

Refrigerant–oil mixtures	Quality	Concentration of sampled oil		
		1%	3%	5%
R-22–mineral oil	0.5	2.02	5.69	9.47
	0.6	2.51	7.02	11.6
	0.7	3.32	9.14	14.8
	0.8	4.90	13.1	20.7
	0.9	9.34	23.2	34.3
R-407C–POE oil	0.5	2.22	5.96	9.36
	0.6	2.75	7.34	11.4
	0.7	3.64	9.55	14.7
	0.8	5.36	13.7	20.5
	0.9	10.1	24.1	34.1

mixed only with the liquid-phase refrigerant during condensation. The local oil concentration for the liquid phase of the mixture, c_{lo}, can be estimated using the follow equation suggested by Eckels et al. [9]:

$$c_{lo} = \frac{c}{(1-c)(1-x)+c}, \tag{1}$$

where x is the quality and c is the sampled oil concentration.

Table 2 shows local oil concentrations for the mixtures of R-22–mineral oil and R-407C–POE oil. The local oil concentration increased significantly with increasing quality for a given sampled oil concentration. Thus, local oil concentration as well as sampled oil concentration should be considered for investigating the effect of refrigeration oil on the condensation heat transfer coefficient. At the same sampled oil concentration, an increase in quality means an increase in local oil concentration for the liquid phase, and the effect of oil on condensation heat transfer becomes significant.

The two test fluids were mixtures of R-22–mineral oil (290 SUS) and R-407C–POE oil (315 SUS). The experimental parameters considered in the present study include the concentrations of injected oil (i.e., 0%, 1%, 3% and 5%), inlet qualities of the test section (0.5–0.9) and mass fluxes (100–400 kg/m^2 s). The pressure at the inlet of the test section was set at 1.5 MPa, and the quality of the refrigerant throughout the test section was controlled to decrease by 0.2 under the present experimental condition.

3. Data reduction and error analysis

The local condensation heat transfer coefficient was obtained by the following equation:

$$h = \frac{q}{T_r - T_{wi}}. \tag{2}$$

Note that heat loss to the surroundings was assumed to be negligibly small. The saturation temperature of the refrigerant was estimated at each thermocouple position of the test section as follows. First, the pressure at each temperature measurement position was estimated from the measured inlet pressure of the test section, the pressure drop across the whole test section and the pressure drop calculated from the Lockhart–Martinelli separated flow model, which was explained in the literature [10]. Chisholm's method in the literature [10] was used for estimating the pressure drop for the two-phase flow in the U-bend. Second, the saturation temperature for R-22 was obtained from its vapor-pressure curve. The saturation temperature for R-407C was obtained from both the saturated pressure and quality calculated from the refrigerant enthalpy at each thermocouple position. Note that R-407C is a ternary zeotropic refrigerant blend of 23% R-32, 25% R-125 and 52% R-134a (by weight), and has a gliding temperature difference of 5.11°C at 1.5 MPa. The saturation temperature of the pure refrigerant was used instead of the saturation temperature of the refrigerant–oil mixture since the temperature difference between the saturation temperature of the refrigerant–oil mixture and that of the pure refrigerant was within the thermocouple resolution under the present experimental condition.

The error analysis for the condensation heat transfer coefficient was carried out using the method suggested by Moffat [11], as shown in the following equation:

$$\frac{\delta h}{h} = \sqrt{\left(\frac{\delta q}{q}\right)^2 + \left(\frac{\delta T_r}{T_r - T_{wi}}\right)^2 + \left(\frac{\delta T_{wi}}{T_r - T_{wi}}\right)^2}. \tag{3}$$

The uncertainty of the condensation heat transfer coefficient was 8.9–10.1% for R-22 and 8.5–9.5% for R-407C.

4. Results and discussions

4.1. Flow pattern in the test section

Fig. 3 shows the present experimental results in a flow pattern map originally proposed by Taitel and Dukler [12].

The Martinelli parameter in Fig. 3, X_{tt}, can be described by the following equation:

$$X_{tt}^2 = \frac{\rho_g}{\rho_f}\left(\frac{1-x}{x}\right)^2 \frac{f_f}{f_g}. \tag{4}$$

The density of the refrigerant–oil mixture was calculated by substituting the quality and local oil concentration of the mixture calculated by Eq. (1) into the equation for the ideal density of a lubricant–refrigerant solution in the ASHRAE Handbook [13]. The mixture viscosity was calculated by substituting the local oil concentration of the mixture into the equation for the viscosity of liquid mixtures suggested by Irving [14], who provided the viscosity of binary-liquid mixtures with a maximum error of 10% in the following equation:

$$\ln \mu_m = y_r \ln \mu_r + y_o \ln \mu_o. \tag{5}$$

The friction factors in Eq. (4) were estimated from a Moody chart by considering the fin height of a microfin tube as the tube roughness. The mixture viscosity increased due to the increase in oil concentration, thus decreasing the Reynolds number. However, the Martinelli parameter, X_{tt}, changed little since the friction factor in the Moody chart did not change much within the experimental Reynolds number range. Fig. 3 shows that the present experimental results fall in the annular flow region of Taitel and Dukler's map. The local oil concentrations did not affect the flow pattern. The Martinelli parameter decreased as the inlet quality increased. As mass flux increased, the factor in the vertical axis of Taitel and Dukler's map, F, increased, but the Martinelli parameter did not vary. The annular flow pattern was experimentally observed through sight glasses located at the inlet and outlet of the present test section, confirming the results given in Fig. 3.

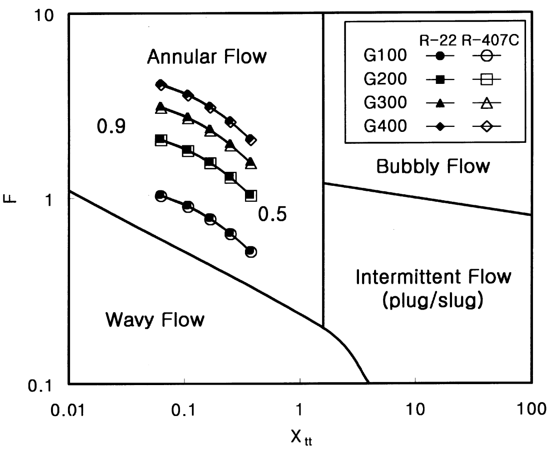

Fig. 3. Taitel and Dukler's flow pattern map for the experimental range.

4.2. Pressure drop in the test section

Fig. 4 shows the predicted and measured pressure drops in the test section. The pressure drop increased as mass flux, inlet quality and oil concentration increased. The pressure drop for R-407C was similar to or slightly lower than that for R-22 because its viscosity was almost identical to that of R-22 within 1.7%. The pressure drops for a sampled oil concentration of 5% were higher by 11–25% for R-22 and 10–20% for R-407C than those for an oil concentration of 0%. The pressure drop between adjacent temperature measuring points was estimated by using the above-mentioned pressure drop data and that in the test section, by the separated flow model modified by Cho and Tae [15] for microfin tubes. The two-phase frictional pressure drop per unit tube length in a microfin tube can be estimated from the liquid-phase frictional pressure drop and two-phase frictional multiplier by considering half of the microfin height as the surface roughness of the tube. Webb [16] proposed 8 for the constant C in the definition of the two-phase frictional multiplier for an enhanced tube like a microfin tube. The frictional pressure drop per unit tube length

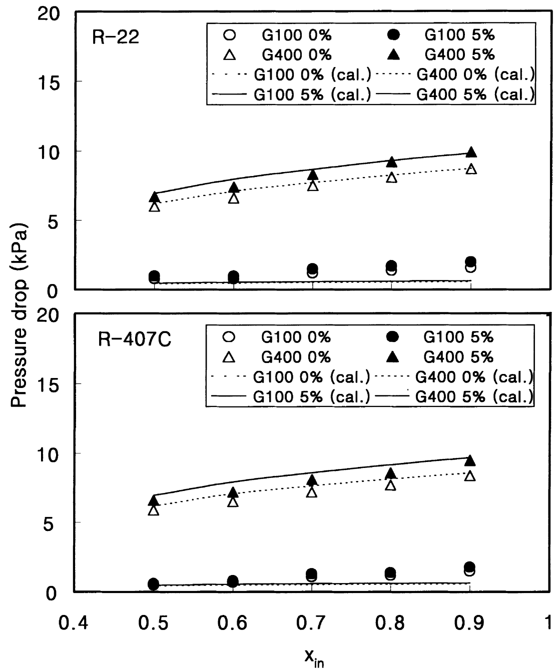

Fig. 4. Pressure drop in the test section.

for a two-phase flow with constant quality can be calculated using the Martinelli parameter. The frictional pressure drop predicted by the separated flow model was estimated by integrating the pressure drop equation with respect to the quality from the inlet to the outlet of the test section. Accelerational pressure drop for two-phase flow was also predicted and added to the frictional pressure drop to obtain the total pressure drop. The accelerational pressure drop was below 13% of the frictional pressure drop within the experimental condition. The pressure drop in the U-bend was estimated using the method suggested by Chisholm in the literature [10]. The pressure drop in the U-bend was within 9% of the total pressure drop. The predicted pressure drops agreed with the measured values within 1.4 kPa for R-22 and 1.2 kPa for R-407C.

4.3. Condensation heat transfer coefficients in the first straight section

Fig. 5 shows the average heat transfer coefficients in the first straight section. The heat transfer coefficients increased as mass flux and inlet quality increased for both R-22 and R-407C refrigerants. However, the heat transfer coefficients for R-22 remained constant as mass flux increased over 300 kg/m^2 s. The reason was that the flow velocity near the inner wall of the microfin tube could not increase due to increasing friction between the inner wall of the microfin tube and the refrigerant as mass flux increased beyond 300 kg/m^2 s. Eckels et al. [5] also reported that condensation heat transfer coefficients for pure refrigerants like R-134a in a microfin tube increased slowly as mass flux increased over 300 kg/m^2 s.

For a refrigerant mixture such as R-407C, the condensation heat transfer coefficients increased continuously as mass flux increased, as shown in Fig. 5. The reason may be explained as follows: for the condensation of a two- or three-component refrigerant mixture, the higher the boiling point of the refrigerant, the sooner it condenses. The difference increases mass transfer resistance at the interface of the liquid and vapor phases and reduces the heat transfer coefficient. The heat transfer coefficients for R-407C were smaller by 12–44% than those for R-22, as shown in Fig. 5. The heat transfer coefficients decreased as the oil concentration increased, even though the heat transfer coefficients increased as mass flux and inlet quality increased for both R-22 and R-407C refrigerant–oil mixtures. Schlager et al. [3] investigated the heat transfer coefficients for R-22 refrigerant–mineral oil (150 and 300 SUS) mixtures in a microfin tube with an outer diameter of 9.52 mm for a mass flux from 125 to 400 kg/m^2 s and the quality from 0.88 to 0.05. Their heat transfer coefficient data for R-22 refrigerant without oil gave fair agreement with the present experimental data within 10% for an inlet quality of 0.5 and oil concentration of 0%, as shown in Fig. 5.

Fig. 6 shows the effect of refrigeration oil on the average heat transfer coefficients at the first straight section in terms of an enhancement factor (EF). EF was defined as the ratio of heat transfer coefficient with oil to that without oil. The EFs for R-22 and R-407C con-

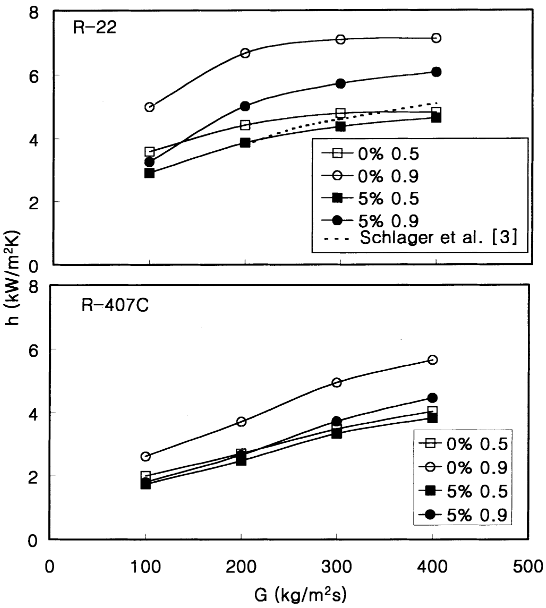

Fig. 5. Average heat transfer coefficients in the first straight section.

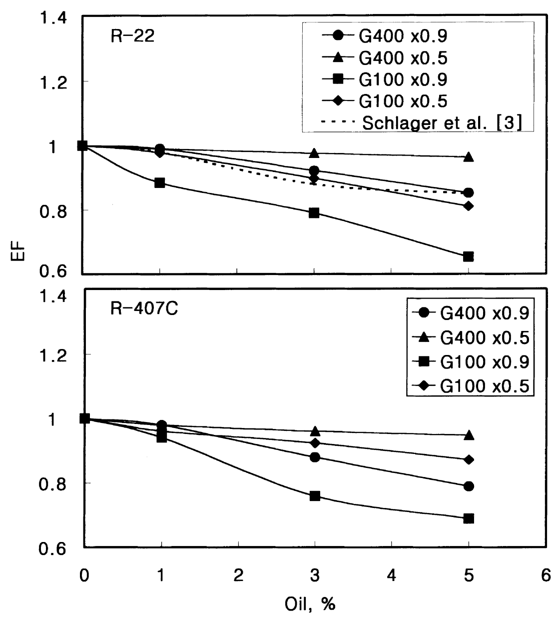

Fig. 6. EF in the first straight section.

tinuously decreased as the oil concentration increased. The minimum value of EF was 0.65 for R-22 and 0.70 for R-407C under conditions of mass flux of 100 kg/m^2 s, inlet quality of 0.9 and oil concentration of 5%. The EF values reported by Schlager et al. [3] for R-22–mineral oil (300 SUS) mixture were compared with the present experimental data in Fig. 6. The degradation of heat transfer coefficient with increasing oil concentration can be explained by the increased liquid-phase viscosity of the refrigerant–oil mixture. The heat transfer coefficients calculated by the Dittus–Boelter equation for a single-phase turbulent flow was found to be proportional to $\mu^{-0.4}$. In the two-phase flow, oil was mixed only with the liquid-phase refrigerant since the vapor pressure of oil was negligibly small compared with that of the refrigerant. Thus, the local oil concentration in the liquid-phase mixture increased with increasing sampled oil concentration and inlet quality, as shown in Table 2, and it makes the viscosity of the liquid phase of the mixture increase. The EFs for a mass flux of 400 kg/m^2 s were larger than those for a mass flux of 100 kg/m^2 s for both refrigerant–oil mixtures. The reason is that the liquid refrigerant and oil were uniformly mixed without forming an oil film as mass flux increased. The EFs for an inlet quality of 0.9 were smaller than those for an inlet quality of 0.5 for both refrigerant–oil mixtures. The reason is that the local oil concentration of the liquid-phase refrigerant–oil mixture increased significantly as the inlet quality increased, as shown in Table 2, and its viscosity increased, thus decreasing the heat transfer coefficients.

4.4. Condensation heat transfer coefficients in the U-bend section

Figs. 7 and 8 show the heat transfer coefficients in the U-bend for R-22 and R-407C, respectively. The heat transfer coefficients in the U-bend were always larger than the average heat transfer coefficients in the first straight section because of flow disturbance due to the centrifugal force. The heat transfer coefficients in the U-bend were at the maximum at the 90° position. For the cases without oil, the heat transfer coefficients for a mass flux of 400 kg/m^2 s were larger than those for a mass flux of 100 kg/m^2 s at the same inlet quality. The heat transfer coefficients for the inlet quality of 0.9 were larger than those for the inlet quality of 0.5. The heat transfer coefficients decreased as the oil concentration increased and decreased rapidly as the inlet quality increased.

The heat transfer coefficients decreased almost at the same magnitude at each point of the U-bend. Therefore, the EFs at the 90° position of the U-bend are typically shown with respect to oil concentration in Fig. 9. The EFs for both R-22 and R-407C decreased rapidly under conditions of mass flux of 100 kg/m^2 s and inlet quality of 0.9. The EFs showed a minimum value of 0.55 for R-22 and 0.63 for R-407C when the oil concentration was 5%. The EFs for a mass flux of 400 kg/m^2 s were larger than those for a mass flux of 100 kg/m^2 s for both refrigerants. The reason is that the flow disturbance due to the centrifugal force in the

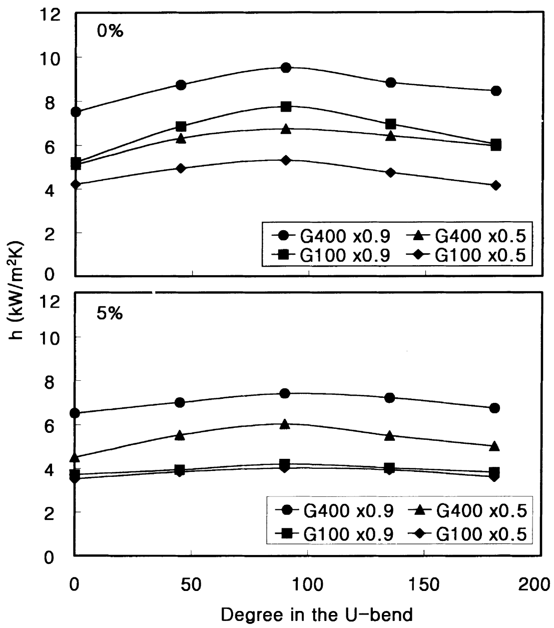

Fig. 7. Heat transfer coefficients in the U-bend section for R-22.

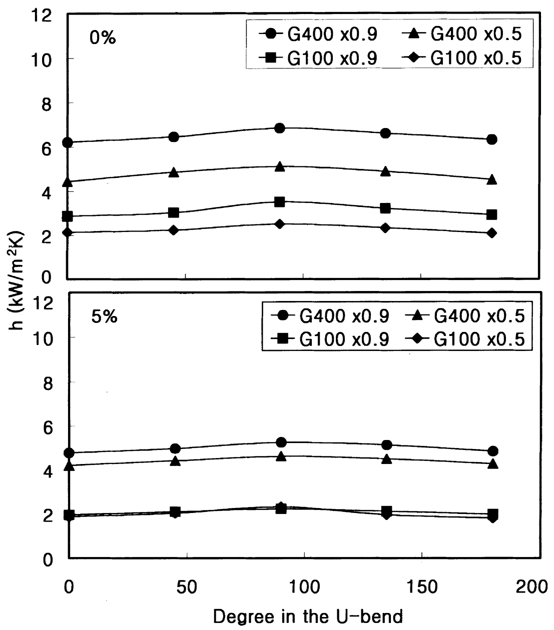

Fig. 8. Heat transfer coefficients in the U-bend section for R-407C.

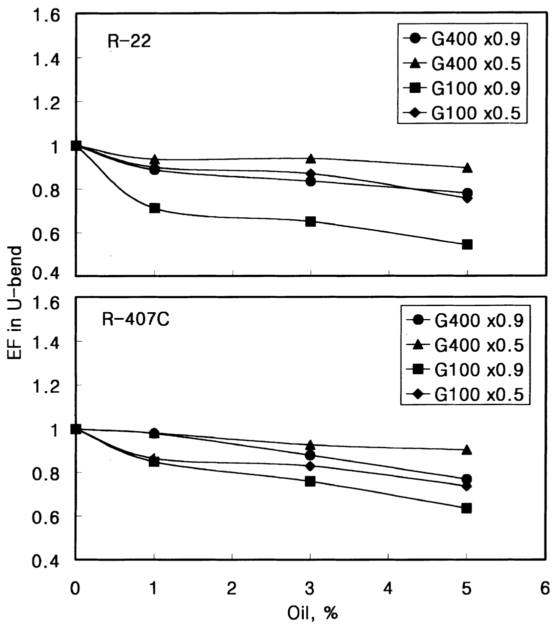

Fig. 9. EF at the 90° position of the U-bend.

U-bend increased as mass flux increased. The EFs, for an inlet quality of 0.9, were smaller than those for one of 0.5, since the local oil concentration in the liquid phase of the refrigerant–oil mixture increased as inlet quality increased.

4.5. Condensation heat transfer coefficients in the second straight section

Fig. 10 shows the dimensionless heat transfer coefficients (h^+) with respect to the dimensionless length (L^+) for both R-22 and R-407C when mass flux was 100 kg/m^2 s and inlet quality was 0.9. The coefficient h^+ is the ratio of the local heat transfer coefficient at the second straight section to the average heat transfer coefficient at the first straight section. L^+ is the ratio of the tube length at the second straight section to the inner tube diameter. The heat transfer coefficients at the second straight section within the dimensionless length of 48 were larger by a maximum of 33% than the average heat transfer coefficients at the first straight section for both R-22 and R-407C refrigerant–oil mixtures. The reason is that the effect of the secondary flow pattern in the U-bend was carried into the downstream straight section after the U-bend. The flow disturbance effect by the U-bend disappeared in the region beyond a dimensionless length of 48. The dimensionless heat transfer coefficients with oil were generally larger than those for pure refrigerants. The reason was that, as oil concentration increased, the heat transfer coefficients near the inlet of the second straight section decreased less than

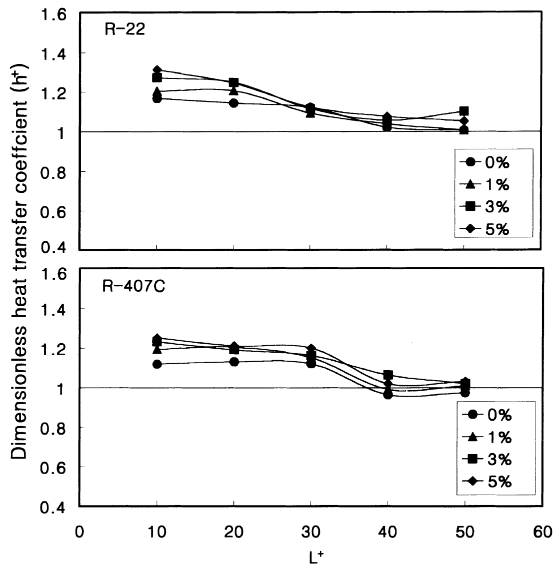

Fig. 10. Dimensionless heat transfer coefficients in the second straight section for a mass flux of 100 kg/m^2 s and inlet quality of 0.9.

those at the first straight section due to the flow disturbance by the U-bend.

Fig. 11 shows the EFs at several positions in the second straight section. The EF was largest at a dimensionless length of 12, the closest position of data collection to the U-bend in the second straight section. The EFs decreased rapidly along the axial direction as

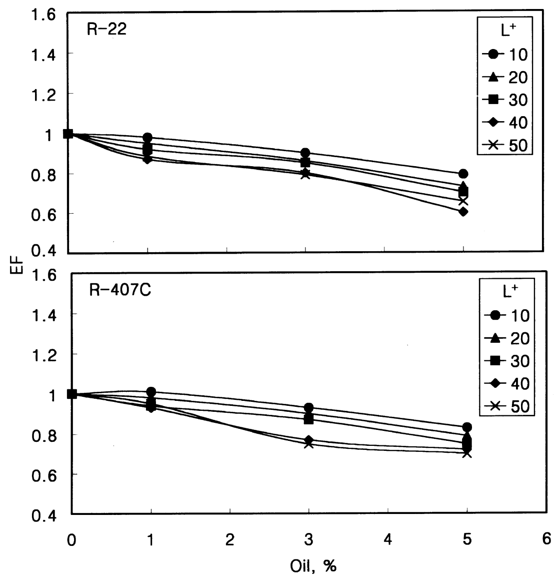

Fig. 11. Enhancement factor at each position of the second straight section for a mass flux of 100 kg/m^2 s and inlet quality of 0.9.

the dimensionless length increased for both R-22 and R-407C refrigerants.

5. Conclusions

The present study conducted experiments to investigate the effect of both POE and mineral oils on the condensation heat transfer behavior for both R-407C and R-22 in a microfin tube with a U-bend. A brief summary is given below:

1. Annular flow pattern was experimentally observed as predicted by Taitel and Dukler's flow pattern map.
2. Pressure drop increased as oil concentration, mass flux and inlet quality increased.
3. The enhancement factors for both R-22 and R-407C refrigerants at the first straight section continuously decreased as oil concentration increased. They decreased rapidly as mass flux decreased and inlet quality increased. The EF had a minimum of 0.65 for R-22 and 0.70 for R-407C for a mass flux of 100 kg/m^2 s, an inlet quality of 0.9 and an oil concentration of 5%.
4. The heat transfer coefficients were maximum at the 90° position of the U-bend. The EFs of both refrigerants decreased rapidly as mass flux decreased and inlet quality increased.
5. The heat transfer coefficients at the second straight section within a dimensionless length of 48 were larger by a maximum of 33% than the average heat transfer coefficient at the first straight section for both R-22 and R-407C refrigerant–oil mixtures, reflecting the flow disturbance effect caused by the U-bend.

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