Effect of Refrigeration Oil on the Condensation Heat Transfer for R-22 and R-407C Refrigerants in Microfin Tube with a U-bend

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Key words: Condensation heat transfer, R-407C, Microfin tube, U-bend, POE oil

Abstract

The present study experimentally investigated the effect of refrigeration oil on the condensation heat transfer for R-22 and R-407C in a microfin tube with a U-bend. Mineral oil and POE oil were used for R-22 and R-407C respectively. Experimental parameters were an oil concentration from 0 to 5%, a mass flux from 100 to 400 kg/m^{2}s and an inlet quality from 0.5 to 0.9. The enhancement factors for both R-22 and R-407C refrigerants at the first straight section decreased continuously as the oil concentration increased. They decreased rapidly as the mass flux decreased and the inlet quality increased. The heat transfer coefficients in the U-bend showed the maximum at the 90° position. 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 coefficient at the first straight section.

Nomenclature

c : oil concentration
EF : enhancement factor
f : Darcy friction factor
G : mass flux [kg/m^{2}s]
h : heat transfer coefficients [kW/m^{2}K]
h^+ : ratio of heat transfer coefficient at second straight section to average heat transfer coefficient at the first straight section
L^+ : ratio of length to inner diameter of tube
q : heat flux in the test section [kW/m^2]
T : temperature [℃]
X_{tt} : Martinelli parameter
x : quality
y : mass fraction

Greek symbols

ρ : density [kg/m^3]
μ : viscosity [kg/m s]

Subscripts

f : liquid phase

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g : gas phase
in : inlet of test section
lo : local
m : refrigerant-oil mixture
o : oil
r : refrigerant
wi : inner wall of test tube

1. Introduction

R-407C, a mixture of R-32/R-125/R-134a, has been recommended as a short-term replacement of R-22 because its properties are similar to R-22. Refrigeration oil is often used for lubricating and sealing of 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 for heat exchanger. Schlager et al.[1] reported that the microfin tube enhanced the heat transfer by 50~100% compared with a smooth tube.

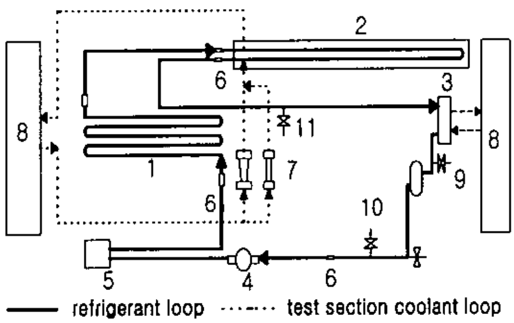
The effect of a refrigeration oil on the condensation heat transfer has been investigated by a number of researchers, who reported the degradation of the condensation heat transfer coefficient due to the presence of the refrigeration oil. Schalager et al.[2-3] and Eckels et al.[4-5] investigated the effect of the refrigeration oil on the condensation heat transfer in a microfin tube. The condensation heat transfer of alternative refrigerant of R-22 was investigated by Jeong et al.[6] and the effect of refrigeration oil on the condensation heat transfer in a smooth tube was reported by Yoon et al.[7]

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. Cho and Tae[8] reported the effect of refrigeration oil on the evaporation heat transfer in a straight and U-bend section of microfin tube. But condensation heat transfer coefficients for a refrigerant-oil mixture was reported only in the straight smooth and microfin tubes. To the authors' knowledge, the effect of refrigeration oils on the condensation heat transfer performance in a microfin tube containing a U-bend has not been reported in the literature.

The objective of the present study was to experimentally investigate the effect of both mineral oil (290 SUS, 62.5 mm^2/s at 40℃) and POE oil (315 SUS, 74.1 mm^2/s at 40℃) on the condensation heat transfer performance for R-22 and R-407C refrigerants in both straight and U-bend sections of a microfin tube.

2. Experimental apparatus and procedure

Figure 1 shows a schematic diagram of the present experimental system, which consists of



1 pre-heater
2 test section
3 plate heat exchanger
4 refrigerant pump
5 mass flow meter
6 sight glass
7 rotameter
8 chiller
9 refrigerant inlet
10 oil inlet
11 sampling port

Fig. 1 Schematic diagram for experimental apparatus.

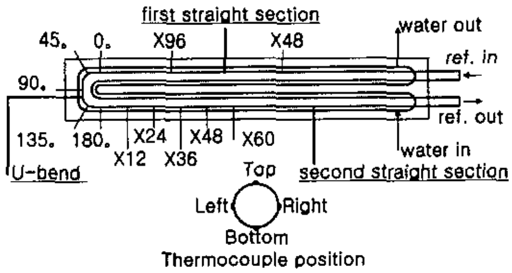


Fig. 2 Details for the test section.

a refrigerant flow loop and two-coolant flow loops. The refrigerant flow loop consists a test section, a pre-heater, a plate heat exchanger, oil injection and sampling devices, a magnetic refrigerant pump, and 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 had 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 was 1.51. The diameter of the curvature for the U-bend was 2.61 times the outer tube diameter.

The outer wall temperature was measured using T-type thermocouple at 12 positions along the direction of refrigerant flow. The thermocouple positions were shown in Fig. 2 and the number following × indicates ratio of tube length from the inlet of the first (or second) straight section to the inner tube diameter for the straight section. Temperature was measured every 45° in the U-bend section of the tube. The outer wall temperature were measured at four circumferential points 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 temperature by applying the radial heat conduction equation for a hollow cylinder.

The coolant temperatures were measured using T-type thermocouples at the inlet and outlet of the test section and the inlet and outlet of the U-bend. The temperature rise of coolant between the inlet and outlet of the test section was approximately 1℃. 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.[9] In order to measure the oil concentration in the test system, the mass of the oil injector and sampling devices before and after the oil injection and boiling-off processes was measured with a balance (4 kg range, 0.1 g resolution).

Table 1 shows measured concentrations of the injected and sampled oil. Since the difference between injected and sampled oil concentration 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.

In this study the two test fluids were mixtures of R-22/mineral oil and R-407C/POE oil. The experimental parameters considered in the present study include the concentration of in-

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

Oil %	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

jected oil (i.e. 0, 1, 3 and 5%), inlet qualities of the test section (0.5~0.9), and mass flux (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 decreased 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.

$$h = \frac{q}{T_r - T_{wi}} \tag{1}$$

Note that the 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] Chisholms 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 the R-22 was obtained from the vapor-pressure curve for the R-22. The saturation temperature for the 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, 52% R-134a (by weight), and has a gliding temperature difference of 5.11℃ 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.

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

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. Result 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]

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

$$X_{tt}^2 = \frac{\rho_g}{\rho_f}\left(\frac{1-x}{x}\right)^2 \frac{f_f}{f_g} \tag{3}$$

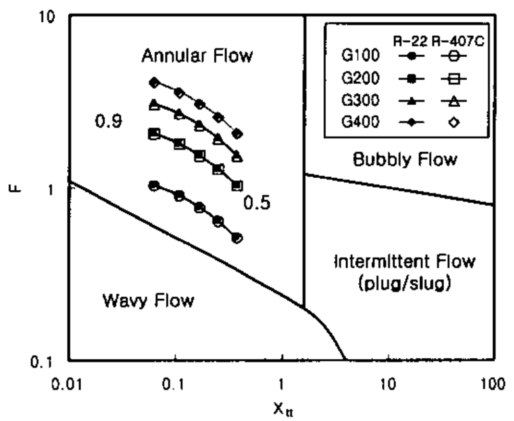


Fig. 3 Taitel and Dukler's flow pattern map for the experimental data.

The density of the refrigerant-oil mixture was calculated by substituting the quality and local oil concentration of the mixture into the equation for the ideal density of a lubricant-refrigerant solution in 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 mixture suggested by Irving.[14] Irving[14] provided the viscosity of binary-liquid mixture with the maximum error of 10% in the following equation.

$$\ln \mu_m = y_r \ln \mu_r + y_o \ln \mu_o \tag{4}$$

Friction factors in equation (3) 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 the 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 to the annular flow region of Taitel and Duklers map. The local oil concentrations did not affect the flow pattern. The Martinelli parameter decreased as the inlet quality increased. As the mass flux increased, the factor in the vertical axis of the Taitel and Duklers map, F, increased but the Martinelli parameter did not vary. 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.

4.2 Condensation heat transfer coefficients in the first straight section

Figure 4 shows the average heat transfer coefficients in the first straight section. The heat transfer coefficients increased as the mass

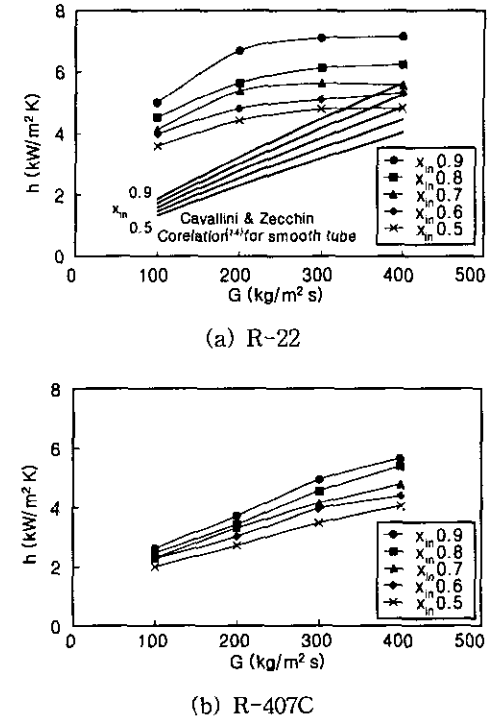
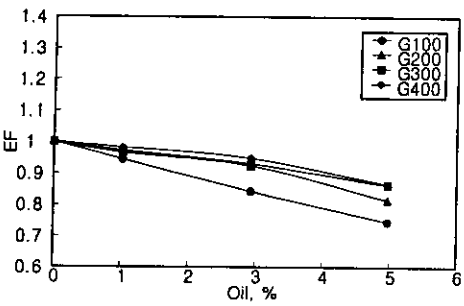


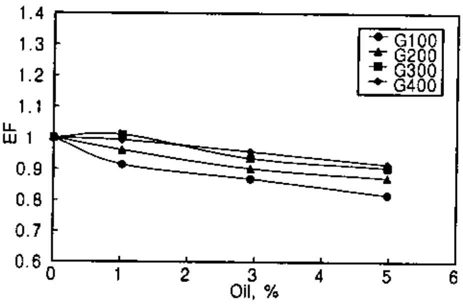
Fig. 4 Heat transfer coefficients in the first straight section without oil.

flux and inlet quality increased for both R-22 and R-407C refrigerants. But the heat transfer coefficients for R-22 remained constant as the mass flux increased over 300 kg/m^{2}s. The reason is that the flow velocity near the inner wall of the microfin tube could not increase due to the increasing friction force between the inner wall of the microfin tube and the refrigerant as the mass flux increased beyond 300 kg/m^{2}s.

For a refrigerant mixture such as R-407C, the condensation heat transfer coefficients increased continuously as the mass flux increased as shown in Fig. 4. The reason may be explained as follows: for the condensation of a two- or three-component refrigerant mixture, the higher boiling point the refrigerant has, the sooner the refrigerant condenses. The difference increases a mass transfer resistance at the interface of liquid and vapor phases, and reduces the heat transfer coefficient. The heat

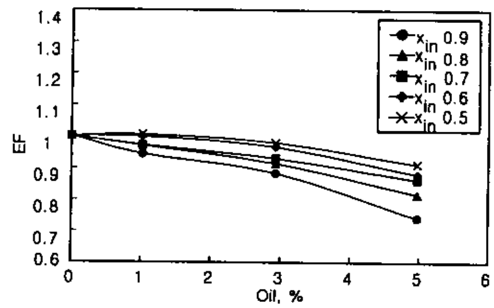


(a) R-22

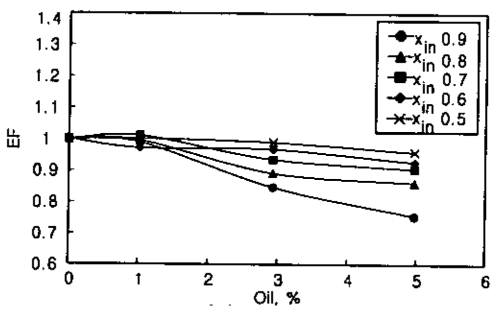


(b) R-407C

Fig. 5 Effect of mass flux on the EFs in the first straight section for the inlet quality of 0.7.



(a) R-22



(b) R-407C

Fig. 6 Effect of inlet quality on the EFs in the first straight section for the mass flux of 300 kg/m^{2}s.

transfer coefficients for R-407C were smaller by 12~44% than those for R-22 as shown in Fig. 4.

Figures 5 and 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). The EF was defined as the ratio of heat transfer coefficient with oil to that without oil. The EFs for R-22 and R-407C continuously decreased as the oil concentration increased. 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 liquid-phase mixture increased with increasing sampled-oil concentration and inlet quality, and it makes the viscosity of liquid-phase of mixture to increase. In Fig. 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 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 as shown in Fig. 6. The reason is that the local oil concentration of liquid-phase refrigerant-oil mixture increased significantly as the inlet quality increased, and the viscosity of liquid-phase refrigerant-oil mixture increased, thus decreasing the heat transfer coefficients.

4.3 Condensation heat transfer coefficient in the U-bend section

Figure 7 shows the heat transfer coefficients in the U-bend for both R-22 and R-407C. The heat transfer coefficients in the U-bend were always larger than the average heat transfer coefficients in the first straight section because of the flow disturbance due to the centrifugal force. The heat transfer coefficients in the U-bend were the maximum at the 90° position. As the oil concentration increased from 0 to 5%, the heat transfer coefficients decreased as in the first straight section. The heat transfer coefficients decreased more rapidly under the condition of mass flux of 100 kg/m^{2}s and inlet quality of 0.9 than the condition of mass flux of 400 kg/m^{2}s and inlet quality of 0.5.

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 were typically shown with respect to the oil concentration in Figs. 8

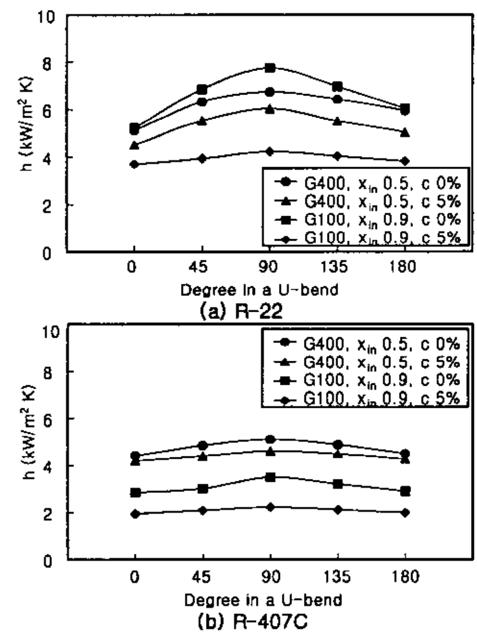


Fig. 7 Heat transfer coefficients in the U-bend.

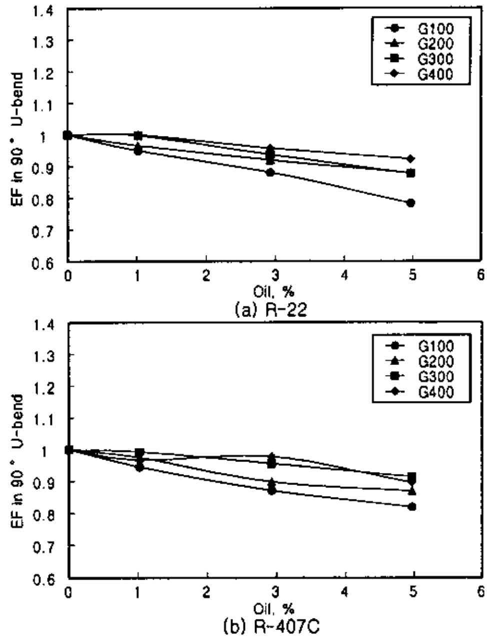


Fig. 8 Effect of the mass flux on the EFs at the 90° position in the U-bend for the inlet quality of 0.7.

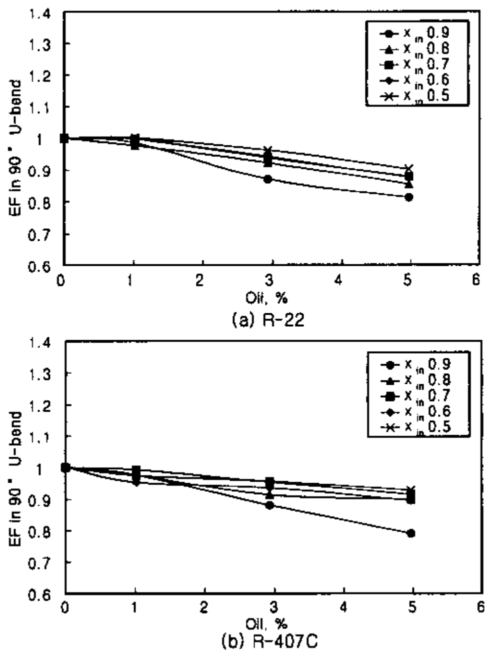


Fig. 9 Effect of the inlet quality on the EFs at the 90° position in the U-bend for the mass flux of 300 kg/m^{2}s.

and 9. Fig. 8 shows the effect of mass flux on the EFs at the 90° position of the U-bend, wheras Fig. 9 shows the effect of inlet quality on the EFs at the 90° position of the U-bend.

In the Fig. 8, The EFs for both R-22 and R-407C decreased rapidly under the condition of lower mass fluxes. The reason is that the flow disturbance due to the centrifugal force in the U-bend increased as the mass flux increased. In the Fig. 9, the EFs for an inlet quality of 0.9 were smaller than those for an inlet quality of 0.5, since the local oil concentration in the liquid phase of the refrigerant-oil mixture increased as the inlet quality increased.

4.4 Condensation heat transfer coefficients in the second straight section

Figure 10 shows the dimensionless heat transfer coefficients (h^+) with respect to the dimensionless length (L^+) for both R-22 and R-407C when the mass flux was 100 kg/m^{2}s and the inlet quality was 0.9. The 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. The L^+ is the ratio of a 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 a 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 those at the first straight section due to the flow disturbance by the U-bend.

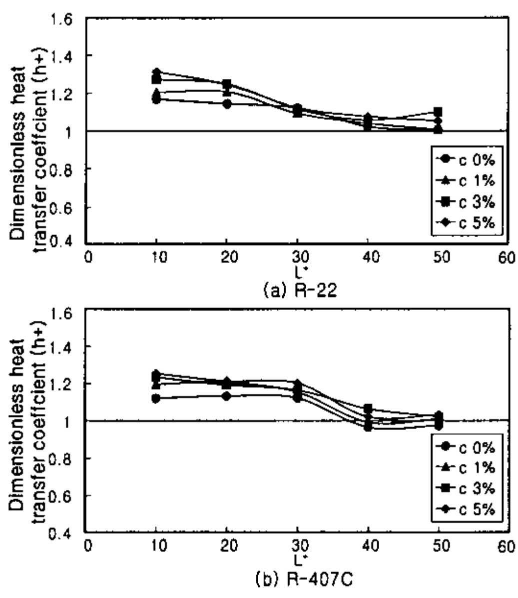


Fig. 10 Dimensionless heat transfer coefficient in the second straight section for the mass flux of 100 kg/m^{2}s and the inlet quality of 0.9.

5. Conclusions

(1) The enhancement factors for both R-22 and R-407C refrigerants at the first straight section continuously decreased as the oil concentration increased. They decreased rapidly as the mass flux decreased and the inlet quality increased.

(2) The heat transfer coefficients had the maximum at the 90° position of the U-bend. The EFs of both refrigerants decreased rapidly as the mass flux decreased and the inlet quality increased.

(3) 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 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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