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Research on condensation heat transfer characteristics of R447A, R1234ze, R134a and R32 in multi-port micro-channel tubes

Minxia Li [a*], Qiang Guo[a], Jiatong Lv[a]

a. Key Laboratory of Efficient Utilization of Low and Medium Grade Energy, MOE, Tianjin University, Tianjin 300350, China

Abstract

The condensation heat transfer characteristics of R447A (R32/R1234ze/R125) in multi-port extruded micro-tubes (MPE) with inner diameter of 0.86 mm were investigated by experimental and compared the results of R134a, R32 and R1234ze.The experiments were conducted under the mass fluxes ranged from 100 to 400 kg/m^2.s,and saturated temperatures from 35 to 45 ^{o}C. The results show that the heat transfer coefficient of R447A was lower than that of R32 greatly and R134a slightly, but higher than R1234ze. On the basis of six revised correlations, the predicted mean absolute errors were within 30% compared with the experimental values.

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Keyword:MPE; convective heat transfer; micro-channel

Introduction

According to the Montreal protocol, CFCs have been prohibited and HCFCs are also being phased out step by step [1]. From the viewpoint of reducing global warming, global warming potential (GWP) of refrigerants from high to low becomes a pressing trend.

Among the low-GWP refrigerants, hydrofluoroolefins (HFOs) such as R32 and R1234ze, and mixtures such as R447A, which have zero ozone depletion potential (ODP) and low GWP, are getting increasing attention as alternatives to R134a and R410A. However, these HFOs have more complex molecule structures than most HFCs [2], and the heat transfer characteristics need to be studied.

Micro-channel heat exchanger as an alternative to traditional exchanger becomes a hot spot in the field of air-condition, especially in automotive field. A new generation of micro-channel heat exchanger can improve the heat transfer performance and reduce the filling amount, in order to achieve the purpose of energy saving and environmental protection, it is significant to understand the heat transfer characteristics of R447A and other low GWP working fluids in micro-channels.

Corresponding author. Tel.: +86 022 27406040; fax: +86 022 27404741.
E-mail addresses: tjmxli@tju.edu.cn (M. Li)

Nomenclature

		out	outlet
A	heat transfer area, m2	sub	sub-cooled
h	specific enthalpy, kJ/kg	*i*	sub-section point
d	diameter, m	s	saturated
d_h	hydraulic diameter	wall	tube wall
d_o	outer diameter	L	Liquid
x	vapor quality	V	Vapor
L	Length , m	1,2,3,4,5,6	State point
Q	evaporation/condensation heat, kJ	cal	calculated
q	Heat flux, kW/m2	exp	experimental

m	mass flow rate, kg/s	**Acronyms**	
Greek symbols		GWP	global warming potential
ζ	interactive coefficient	ODP	ozone depletion potential
δ	wall thickness, m	MPE	multi-port extruded
λ	thermal conductivity, W/m.k	CFCs	chlorofluorocarbons
Subscripts		HCFCs	hydrochloroflurocarbons
r	refrigerant	HFOs	hydrofluoroolefins
w	water	MAE	mean absolute error
in	inlet	HVAC	Heating ventilation and air conditioning

1. Cycle analysis

To analyze the performance of refrigeration cycle system using mixture working fluids, it needs to be compared with the pure refrigerants in the mixture. A performance comparison with R410A which is widely used presently is also needed. The selected working conditions are shown in Table 1:

Table 1. Operating parameters of refrigerant cycle system

Evaporation temperature	7.2 ℃
Condensation temperature	54.4 ℃
Super-heat	11.1 ℃
Sub-cooling	8.3 ℃
Compressor efficiency	85%

Note that the throttling process is isenthalpic throttling. Fig.1 shows the diagram of system cycle, which is summarized as follows: 6-0 is isobaric evaporation process in the evaporator, if the refrigerant is non-azeotropic refrigerants, temperature glide exists in evaporator, the selected evaporation temperature is the mathematical average temperature between point 6 and point 0 , after 0-1 process , the mixture is sucked into the compressor, after 1-2 isentropic compression process, condensed in cooling device (2-3-4). For non-azeotropic refrigerant, the selected condensation temperature is the mathematical average temperature between point 3 and point 4. After the sub-cooling process (4-5), the refrigerants pass through the isenthalpic throttling process (5-6), completing a cycle.

Table 2 shows the calculated results of COP, condensation temperature and exhaust pressure based on the parameters shown in Table 1. The COP of R447A was 3.839, which was higher than that of R410A, furthermore, specific cooling capacity was slightly lower than R32, and the exhaust temperature was slightly higher than R410A. This was due to the low boiling point of R1234ze, which reduced exhaust pressure and ensures the reliability of the compressor. By the contrast between t_3 and t_4 in Fig.1, the temperature glide of R447A was 3-5°C larger than that of R410A. Overall, the ternary mixture of R447A was an alternative to high-GWP refrigerant of R410A.

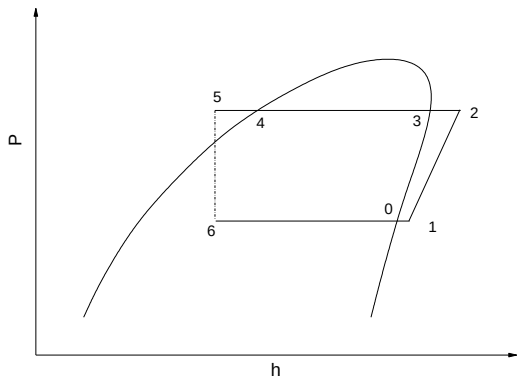


Fig.1. Refrigeration cycle diagram

Table 2. different refrigerant condensation temperature contrast and theoretical COP

Refrigerants	Specific cooling capacity q（kJ/m³)	Cooling coefficient (COP)	Exhaust temperature t_2（°C）	Saturation vapor temperature t_3（°C）	Saturation liquid temperature t_4（°C）	Exhaust pressure p_2（MPa）
R447A	5056.112	3.839	93.149	56.190	52.697	2.934
R410A	5653.579	3.767	84.936	54.435	54.350	3.389
R1234ze	1927.961	4.202	62.354	54.4	54.4	1.114
R32	6236.210	3.853	101.04	54.4	54.4	3.473

2 Condensation heat transfer test

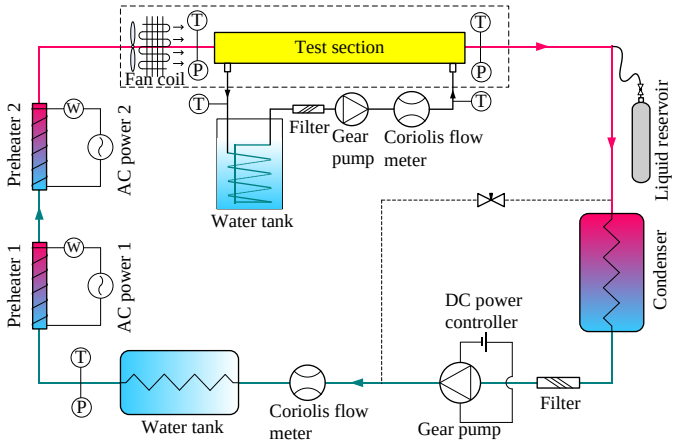


Fig.2. Schematic of experimental setup

The MPE with circular aperture was tested in this paper. The dimension of cross sections were 16*1.3 mm. The effective heating length was 480 mm, and twelve T-type thermo-couples, whose diameter was 0.13mm, welded to the six cross sections. And there was a thermo-couple on the upper and lower outer wall surface of each section to measure the temperature of the wall. Besides, ethanol was taken as the heat transfer fluid in this test, flowing in the 2 mm slit between the flat tube and the shell. MPE was circular aperture tube with 15 holes, whose diameter was 0.86 mm and thickness was 0.22 mm.

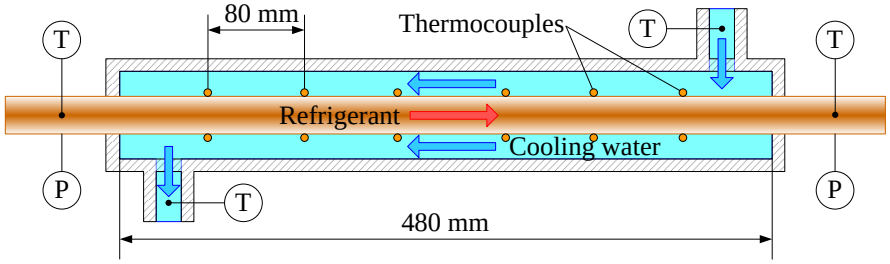


Fig. 3. Test section of MPE

Set the saturated temperatures of 35 °C, 40 °C and 45 °C and the mass fluxes of 100-300 kg/(m²·s). The condensation heat transfer coefficients of the six mixed refrigerants were obtained in the experiment under different conditions, and the effect of vapor quality, mass flux and condensation temperature on heat transfer coefficient were analyzed.

3. Data processing

The refrigerant was cooled to a sub-cooled state by cooling water before preheating, the super-heated temperature was 1-2 °C by electric heating in preheating section. The entrance vapor quality was calculated by Eq.(1).

$$x_{\text{in}} = \left(h_{\text{r, in}} - h_{\text{L}}\right) \Big/ \left(h_{\text{V}} - h_{\text{L}}\right) \tag{1}$$

Where x_{in} is the vapor quality of tube inlet, and $h_{\text{r,in}}$, h_{L} and h_{V} are specific enthalpy of the inlet, liquid phase and gas phase.

Q_{water} is heat flux on water side in the heat exchanger, which can be obtained from the mass flux m_{water}, specific enthalpy $C_{p,water}$, and the inlet and outlet temperature s $T_{\text{out,water}}$ and $T_{\text{in, water}}$ by Eq.(2)

$$Q_{\text{water}} = m_{\text{water}} c_{p,\text{ water}} \left(T_{\text{out, water}} - T_{\text{in, water}}\right) \tag{2}$$

Outlet vapor quality x_{out} was calculated from the inlet vapor quality x_{in} through Eq.(3) based on thermal equilibrium of water side and refrigerant side.

$$x_{\text{out}} = x_{\text{in}} - Q_{\text{water}} \Big/ \left[m_{\text{r}} \left(h_{\text{V}} - h_{\text{L}}\right) \right] \tag{3}$$

The vapor quality of each test point in test section can be obtained by Eq.(4).

$$x_i = x_{\text{in}} - \left(x_{\text{in}} \text{-} x_{\text{out}}\right) \frac{\text{L}_i}{\text{L}} \tag{4}$$

Where L is the total length of effective heat exchange, L_i is the length of test segment.

The heat flux is the average value of heat exchanger segment, determined by Eq.(5) .

$$q = Q_{\text{water}} / A_{\text{h}} \tag{5}$$

Where A_{h} is the heat transfer area of each test segment.

The average convective heat transfer coefficient for each test segment was obtained by Eq.(6).

$$h_i = q / (T_s - T_{wall,in,i}) \tag{6}$$

Where T_{s} is the saturated temperature corresponding to saturated pressure. $T_{\text{wall, in, } i}$ is the temperature of the inner wall surface, which is calculated by Eq.(7) based on the average outer wall temperature of each subsection in the test section.

$$T_{\text{wall, in, } i} = T_{\text{wall, out, } i} + \frac{q}{\lambda_{tube}} \delta \tag{7}$$

Where $T_{\text{wall,out, } i}$ is the average temperature of the outer wall , which is the average value of the measurements of four thermo-couples on outer wall, and δ and λ_{tube} are the thickness and thermal conductivity of MPE.

The mean absolute error (MAE) was calculated by Eq.(8).

$$\text{MAE} = \frac{1}{n} \sum_{1}^{n} \frac{\left|R_{\text{exp}} - R_{\text{cal}}\right|}{R_{\text{exp}}} \tag{8}$$

4. Results and analysis of condensation test

The saturated temperature was 35°C with the mass flow rate of 100-300 kg/(m²·s), the condensation heat transfer coefficients of the six mixed refrigerants were obtained in the experiment under different conditions.

The new low-GWP ternary refrigerant R447A was studied for condensation heat ex-changer. Similar with pure working fluid, the increasing vapor quality will lead to the thinning of film thickness, which will increase the heat transfer coefficient, and with the increasing mass flux, the Reynolds number and disturbance raise, which will also improve the heat transfer coefficient. Because the shear stress plays a leading role at large mass flow, which decreases the mass transfer deterioration, and the influence of the mass transfer resistance at small mass flow is more significant. As Fig.4 shows, similar with other refrigerants, the condensation heat transfer coefficient of R447A increased with the improvement of mass flow and vapor quality. The convective heat transfer coefficients corresponding to the condensation temperatures of 35 °C at the mass flux of 100-300 kg/(m².s) are shown in Fig.4.

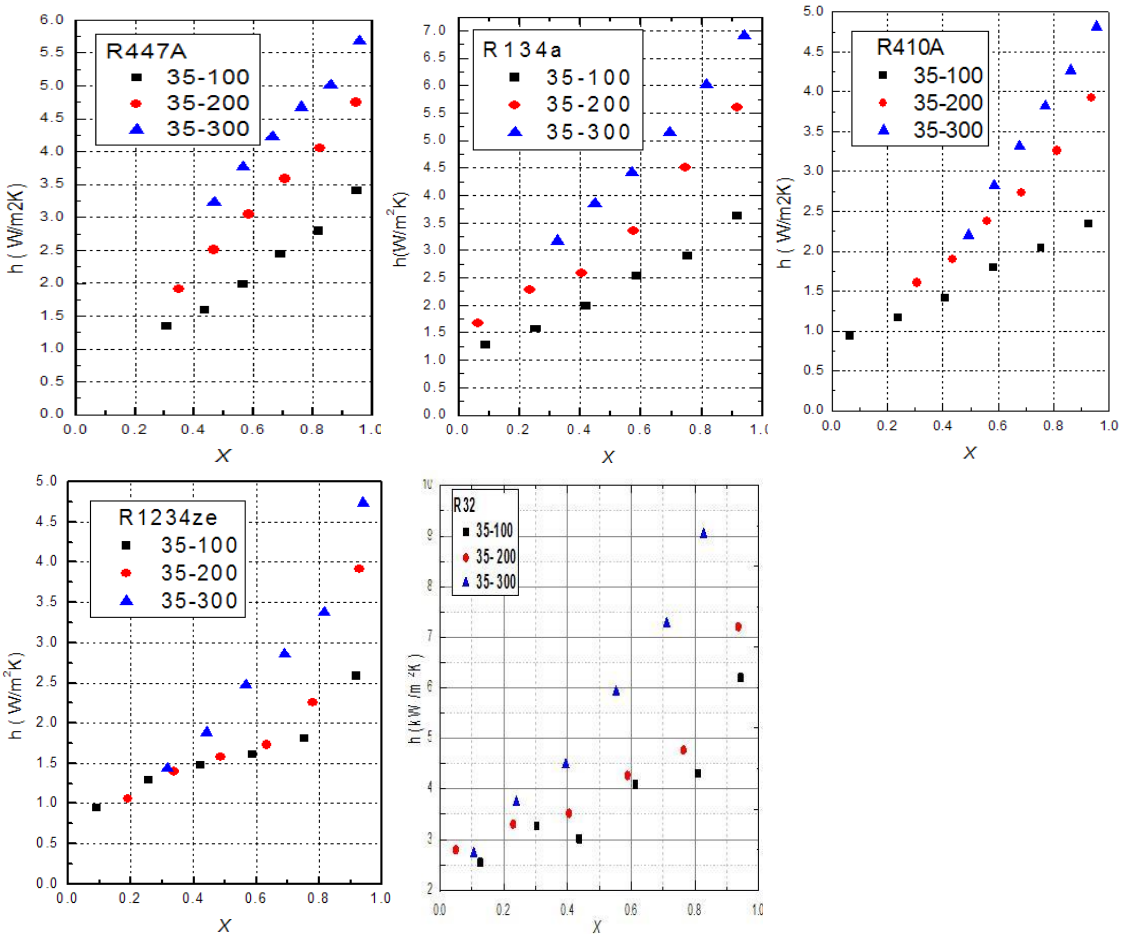


Fig.4. Results of convective heat transfer coefficients of different refrigerants

The heat transfer coefficients of R32/R134a (24.5%/75.5%) and R32/R1234ze (45%/55%) in 0.86mm MPE are shown in Fig.5. Similar with the case of pure refrigerants, with the increase of vapor quality, the film thickness become thinner, leading to higher heat transfer coefficients. The increasing mass flux raised Reynolds number and disturbance, results in the improvement of heat transfer performance. It was noted that the increase rate of heat transfer coefficients slow down at the mass flux of 200 kg/(m².s) under higher vapor quality. By forecasting the flow patterns, the flow patterns were wavy flow under any vapor quality at the mass flux of 100 kg/(m².s). At the mass flux of 200 kg/(m².s), it was intermittent flow under low vapor quality and annular flow under high vapor quality. When the mass flux was 300 kg/(m2.s), all the flow patterns were annular flow. For pure working fluids, the heat transfer coefficient under annular flow was higher than that of intermittent flow. But for mixed working fluids, because of the existence of mass transfer resistance, there existed the variety of heat transfer coefficient with the change of flow pattern at the mass flux of 200 kg/(m²·s). But it was annular flow at the mass flux of 300 kg/(m²·s), which was a relative high mass flux. It means that the mass transfer resistance is not clear

at a high mass flux than that of a low mass flux, shear stress plays a main role at high mass flux, leading to the mass transfer deterioration.

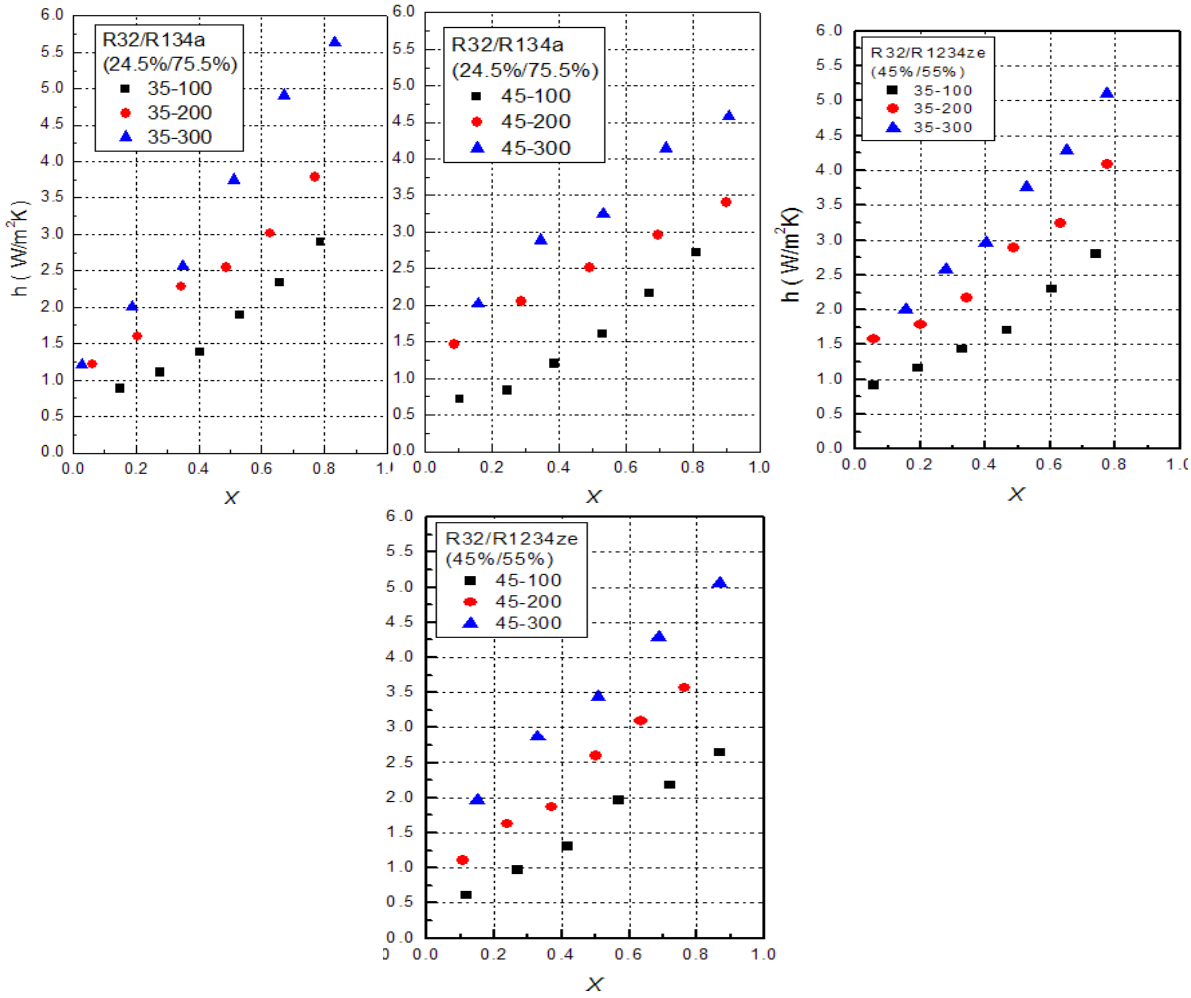


Fig.5. Results of convective heat transfer coefficients of mixed refrigerants

5. Heat transfer coefficient of the theoretical predictions

The condensation heat transfer coefficients were predicted based on the 324 operating points in the test. Besides, the absolute errors were analyzed based on the correlations recommended by Park[3], Cavallini[4], Thome[5], Shah[6] Wang[7], Kim[8], in which the four equations of Park[3], Cavallini[7], Wang[23],Kim[8] are recommended for condensation prediction in MPE. As it is shown in Fig.6, based on the revised SBG balance method, the predicted results of R410A and other binaries of the six correlations were within 30% compared with the experimental results. For R410A, the accuracy of Park[3] correlation was better than any other correlation. For the two mixtures of R32/R134a(76%/24%), the Cavallini[4] correlation had the best accuracy. The predicted accuracy of Thome[5] correlation was better for the two mixtures of R32/R1234ze (45%/55%), and the mean errors was 15.83%. The best predicted model for R447A is Shah[6] correlation.

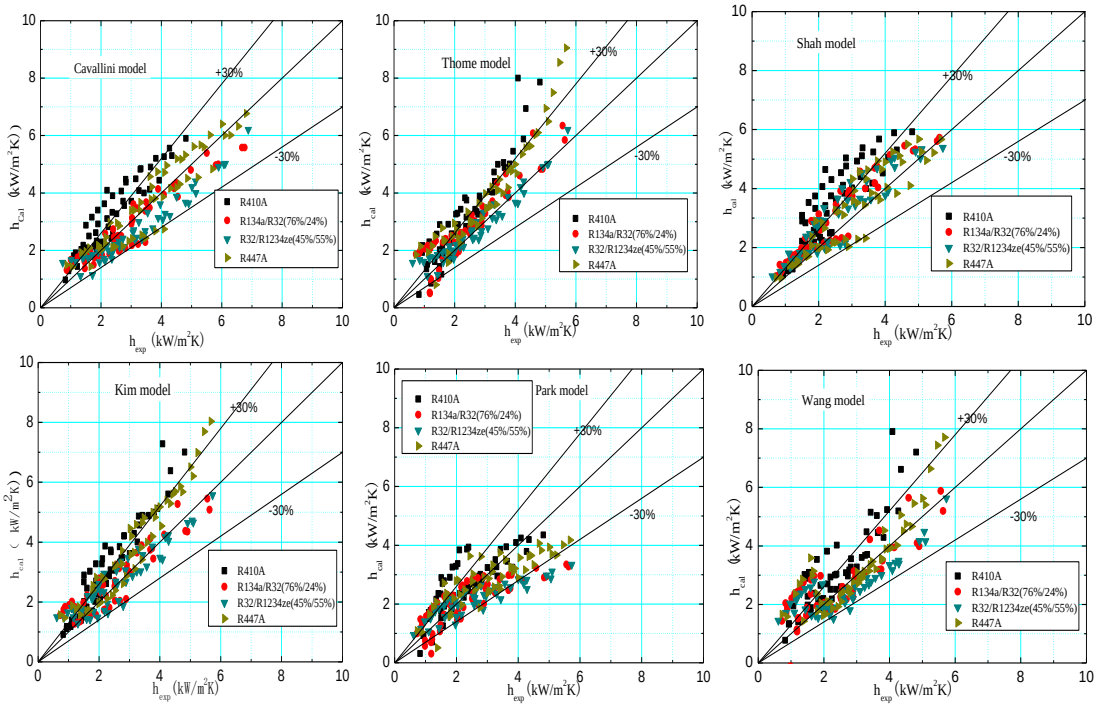


Fig.6. Predicted results of condensation heat transfer correlations for mixtures

6. Conclusion

Predictions and analysis were performed based on the experiments of flow condensation heat transfer for R32/R134a (24%/76% by mass), R32/R1234ze (45%/55% by mass), R410A and R447A in 0.86mm circle multi-port multi-channel tube. The results are summarized as follows:

(1) For R32/R134a (24%/76%) at the mass flux of 200 kg/(m2.s), with the decreasing vapor quality. Because the mass transfer resistance greatly weak the heat transfer coefficient of annular flow, slowing down the increasing trend of condensation heat transfer coefficient at high vapor quality.

(2) By making a comprehensive comparison of all experiments' tested refrigerants, sorting the condensation heat transfer coefficients under the same condition as:

R32>R134a >R447A> R32/R1234ze (45%/55%)>R32/R134a (24.5%/76.5%)>R410A>R1234ze.

(3) The predicted results of the six correlations are within 30% for binary mixtures. For R410A, the accuracy of Park[3] correlation is better than any other correlation. For the mixtures of R32/R134a(76%/24%), the Cavallini[4] correlation have the best accuracy. The best predicted correlation for R447A is Shah[6] correlation. The predicted accuracy of Thome[5] correlation is better for the of R32/R1234ze(45%/55%).

Acknowledgments

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