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Experimental investigation of heat transfer coefficient of R1234yf during condensation inside multiport mini-channel tube

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Abstract

R1234yf is an Hydrofluoro-Olefins (HFO) that is used as a good candidate for replacing R134a in automotive air conditioning, and it has a low global warming potential of 4 and zero ozone depletion potential (ODP). R1234yf has cooling performance that is 5 % less than R134A but is significantly better for the environment. This research report measured and analyzed the heat transfer performances of R1234yf in the condensation process inside two different multiport mini-channel tubes. Experimental data was obtained for mass fluxes from (50 to 500) kg/m²s, and heat fluxes from (3 to 12) kW/m² at a fixed saturation temperature of 48 °C. The flow distribution of the R1234yf effect on the heat transfer coefficient was analyzed. Finally, a new heat transfer coefficient correlation was developed from the experimental data.

Keywords: R1234yf, heat transfer coefficient, condensation, multiport tube.

1. Main text

The R-1234yf refrigerant is a low global warming potential of 4 and zero ODP; it has physical properties similar to R-134A and also has the potential as a retrofit refrigerant in existing HFC-134a systems. R-1234yf is classified as slightly flammable., so it is carefully to ensure safety during operation and maintenance of the system included that. R-1234yf has a cooling performance 5% less than R-134A, but significantly better for the environment.

Many available experimental studies compare the heat transfer performance between R134a and R1234yf. Yang and Nalbandian [1] measured the condensation heat transfer and pressure drop of R1234yf and R134a in a 4 mm inner diameter (ID) circular tube. They found that for high vapor qualities, the heat transfer coefficients of R134a are higher than those of R1234yf, but for low vapor qualities, almost the same. Moreover, the authors concluded that the effect of two-phase flow patterns shows strongly on the condensation heat transfer coefficient. [2]'s experimental study on the condensation of R1234yf inside 0.96 mm circular tube compared it with that of R134a at the same test conditions. They found that R134a showed higher performance than R1234yf by about (15–30) % of heat transfer coefficients, and (10–12) % of adiabatic two-phase pressure drop. The experimental data were compared with the correlation of Cavallini et al.[3] . Wang et al. [4] also measured the condensation heat transfer coefficient of R1234yf inside a 4 mm ID tube, compared it with that of R134a and R32, and concluded that strong effects of the mass flux and vapor quality on the heat transfer coefficient are observed in the shear force dominated flow regimes. Also, due to the differences in density ratio, viscosity ratio, and thermal conductivity, the heat transfer coefficients of R1234yf are lower than those of R134a. Illán-Gómez et al. [5] investigated the condensation heat transfer coefficient and pressure drop inside a multiport tube with R1234yf and R134a with large ranges of mass flux and saturation temperature. They

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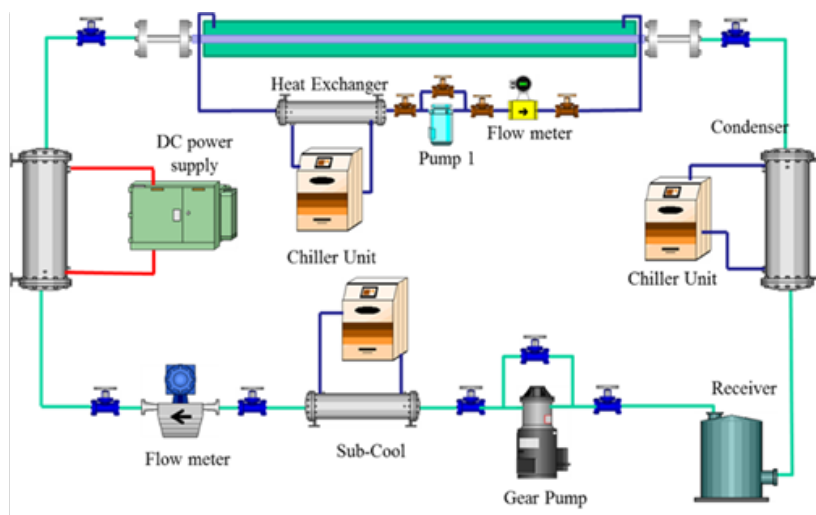


Figure 1 Schematic of the experimental apparatus

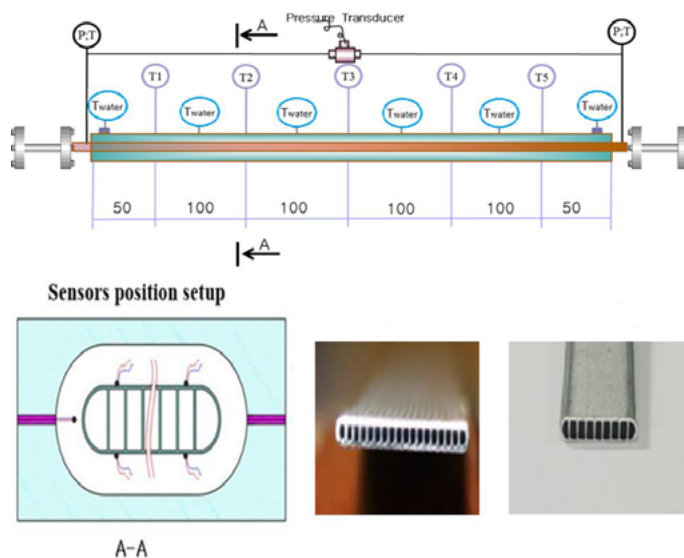


Figure 2 Detail of the test sections setup

compared the experimental data with some heat transfer coefficient and pressure drop models available in the literature, which showed good agreement. Based on the model proposed by Koyama et al. [6], a readjusted model for the heat transfer coefficient has been proposed to improve the agreement with the original model.

Much of the previous research has been conducted and models developed for limited refrigerants and test tubes, while there is a lack of research in the literature that pays attention to the R1234yf inside the multiport minichannel tube. In this study, the condensation heat transfer coefficients of R1234yf inside two types of multiport minichannel tubes of (0.83 and 0.97) mm of hydraulic diameter are investigated experimentally. The effect of mass flux, heat flux, vapor quality, and the geometry of the test section are analyzed, and compared with some existing models.

Table 1 Specification dimensions of the multiport mini-channel tubes

Parameters	PFC	FME
Number of channels	18	8
Width of channel (mm)	0.58	0.69
Height of channel (mm)	1.3	1.42
Hydraulic diameter (mm)	0.846	0.969
Internal cross- sectional area (mm ²)	28.105	19.46
Wetted perimeter surface (mm)	64.52	36.12

2. Experimental apparatus and data reduction

Figure 1 shows the experimental apparatus, which consists of a refrigerant loop, cooling water loop, and water–glycol loop. The experimental apparatus was designed to measure the heat transfer coefficient and pressure drop during the condensation process of refrigerant inside the test section tubes. The refrigerant loop is composed of four major sections: the sub-cooled, the pre-heater, the test section, and the post-condensers. The sub-cooled includes a sub-cooler, a refrigerant gear pump, and a Coriolis-type mass flow meter. The sub-cooler is a tube-in-tube heat exchanger with refrigerant flow inside of the tube and a cooling water loop; the refrigerant is cooled by a water–glycol loop to achieve a subcooled state. The subcooled temperature of refrigerants is controlled by setting the temperatures of constant temperature baths connected to the sub-cooler. The refrigerant is circulated in the refrigerant loop by a magnetic micro gear pump. A Coriolis-type mass flow meter for a maximum mass flow rate of 80 kg/h is installed to measure the mass flow rate of refrigerant. The mass flow rate of refrigerant is independently controlled by adjusting the rotation speed of the drive for the gear pump. The aims of the sub-cooled part provided for the subcooled state refrigerant and required mass flow rate. The pre-heater is used for the multiport tubes test, which is 4 m long, and of 1/4" outer diameter (OD) stainless steel tube. It is connected with a DC power supply for a maximum capacity of 1,500 W. The refrigerant vapor quality in the test section is determined by the heat applied in the pre-heater, which can be controlled by adjusting the voltage and current in the DC power supply unit.

Figure 2 details the test section setup and test section tubes. The test sections are the tube-in-tube heat exchanger in which the refrigerant vapor condenses inside the inner test tubes, while cooling water flows in the annulus. Two types of aluminum multiports were used to have 8 and 16 rectangular channels in hydraulic diameters of (0.969 and 0.846) mm, respectively. The effective lengths of the test section are 200 mm with mass fluxes of (50, 100, and 200) kg/m²s; and 500 mm with mass fluxes of (300 and 500) kg/m²s. The local heat transfer coefficients are calculated from the measurements of average heat flux on the test section and the local tube wall temperatures. Twenty T-type thermocouples were equispaced at fixed 5 points along the test sections length, on both the top and bottom sides, to measure the outside tube wall temperature.

Four high-accuracy resistance temperature detectors were attached at the inlet and outlet of refrigerant side and cooling waterside in the test section. The energy balance in the test section is verified by initial testing with the liquid water–water in the test section; the heat balance between the outside fluid and inside fluid was always less than ±3 %. Table 1 shows the dimensions of the two types of aluminum multiport tube.

The vapor quality at the inlet of the test section was defined from the voltage and current of the DC power supply unit in the preheater, which can be determined as follows:

$$x_{inlet} = \frac{1}{h_{fg}} \left[\frac{Q_{pre-heater}}{m_{ref}} - C_p (T_{sat} - T_{p,inlet}) \right] \quad (1)$$

where, Q_{pre} is the heating power applied in the pre-heater by the DC power supply. The saturation temperature T_{sat} was computed from the measured saturation pressure at the inlet of the test section. The average heat flux in the test section was determined as:

$$q = \frac{m_{water} c_{p,water} \Delta T_{water}}{A_{external}} \quad (2)$$

where, m_{water} is the cooling water mass flow rate, T_{water} is the temperature difference between the inlet and outlet of the cooling water side, and $A_{external}$ is the external surface area of the test section tube. The cooling water mass flow rate in all cases is fixed at 1 kg/min to maintain the water flow distribution on the annulus side. Two test section lengths were used with different mass flux conditions to control the vapor quality difference in the test section at approximately (0.011–0.16). Therefore, it can be assumed that the variation in vapor quality on the test section is linear over the length of the test section. The following equation calculates the vapor quality in the test section:

$$x = x_{inlet} - \frac{m_{water} c_{p,water} \Delta T_{water}}{2m_{ref} h_{fg}} \quad (3)$$

The Fourier steady-state one-dimensional heat conduction calculated the inner wall temperatures of the tube through the tube wall:

$$T_{w,i} = T_{w,o} + \frac{q\delta}{k_{aluminum}} \quad (4)$$

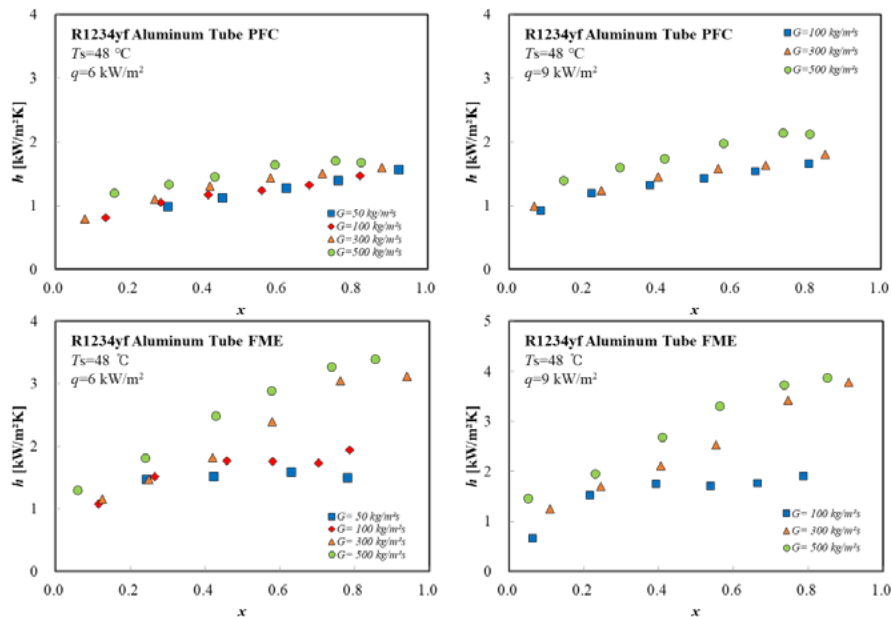


Figure 3 Effect of mass flux on the heat transfer coefficient

where, k is the thermal conductivity of the aluminum. Through all the data reduction processing, the different temperature between inside and outside wall tubes was less than 0.025 K.

The average condensation heat transfer coefficient is obtained as:

$$h = \frac{A_{\text{external}}}{A_{\text{internal}}} \frac{q}{(T_{\text{sat}} - T_{w,i})} \quad (5)$$

3. Result and discussion

Condensation heat transfer of R1234yf inside the aluminum multiport tubes is carried out at fixed 48 °C, mass flux ranging (50 to 500) kg/m²s, and heat flux ranging (3 to 12) kW/m². The vapor quality at the inlet of the test section varied from (0.9 to 0.1), and approximately (0.011–0.16) vapor quality change in the test section, depending on the mass flux and heat flux conditions. The relations of vapor quality, mass flux, and heat flux with the condensation heat transfer coefficient of refrigerant were discussed.

Figure 3 shows the heat transfer coefficient influence of mass flux with both test tubes. The increase of vapor quality and mass flux leads to the rise of the condensation heat transfer coefficient, because of the forced convection mechanism. The effect of vapor quality on the condensation heat transfer coefficient can be explained by flow models. The flow models proposed by Coleman and Garimella [7] show bubbly flow and plug flow in the case of low vapor quality, and annular flow for high vapor quality. Liquid film during condensation is thick in the bubbly and plug flow, and thin in the annular flow. In addition, the effect of liquid film thermal resistance predominates in the condensation heat transfer coefficient, where the heat transfer coefficient increases as the thickness of the liquid film decreases. This effect is more evident in high mass flux and high vapor quality, but weaker in low mass flux and vapor quality.

At low mass fluxes, the results showed that the data of (50, 100, and 150) kg/m²s mass fluxes almost overlap with those figures of (3–12) kW/m² of heat flux. Jige et al. [8], Matkovic et al. [9], and Del Col et al. [10] reported that the heat transfer coefficients of (100 and 200) kg/m²s mass velocity were almost similar, which may be explained by the decrease in the shear stress at a certain point and increase at the interface, so that the effect of surface tension in the rectangular channels dominates. The explanation is similar to the theoretical analysis reported by Bortolin et al. [11]. Kim & Shin [12] considered the condensation heat transfer of R134a inside mini channel tubes, and concluded that the low mass flux regime had a weak effect on the heat transfer coefficient. However, at high mass fluxes, the surface tension does not dominate, because the vapor shear stress is much stronger.

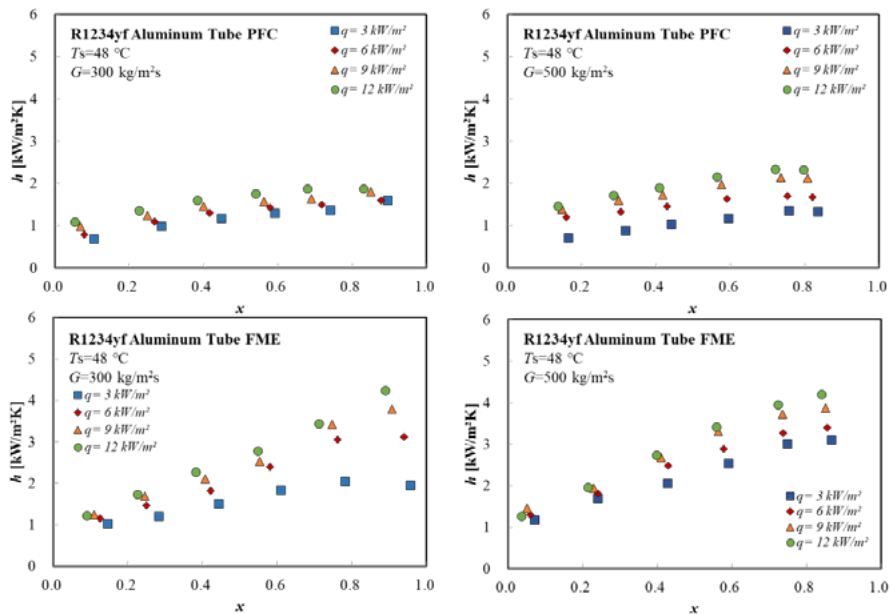


Figure 4 Effect of heat flux on the heat transfer coefficient

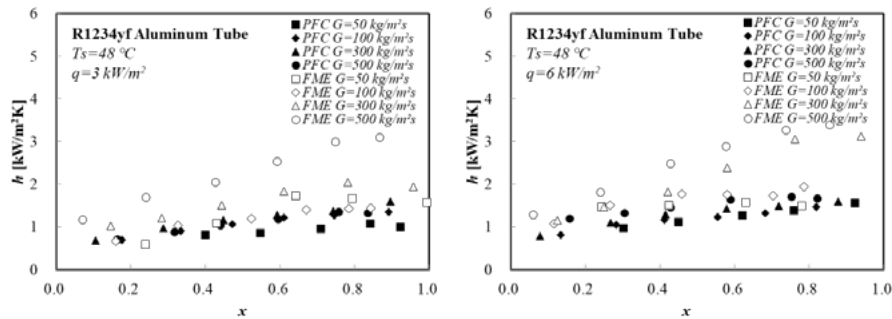


Figure 5 Comparison of the heat transfer coefficient between two types of multiport tubes

Figure 4 shows the effects of heat flux on the measured condensation heat transfer coefficient. The results show that the condensation heat transfer coefficients increase with increasing heat flux. In the present experimental study, the average heat flux approximated uniform heat flux on the test section length, due to keeping 1 kg/min past the cooling water mass flow rate for all test cases. Zhang et al. [13], Sakamatapan et al. [14], and Yan and Lin [15] also reported similar behavior of the heat flux effect. They concluded that the main reasons are the interface temperature between the liquid film and vapor core, and even the effects of the change of momentum of the liquid and vapor phases on the change of heat flux.

Figure 5 compares the condensation heat transfer coefficients of the two types of multiport tube. These show that the heat transfer coefficients of FME are higher than those of PFC at similar test conditions. For conventional round tubes, the heat transfer coefficient increases with decrease of hydraulic diameter of the tubes, but many researchers, such as Pham et al. [16], Kaew-On et al. [17], and Nema et al. [18], suggest that the heat transfer behavior inside multiport rectangular channel tubes is quite different. In the present study, the experimental data show that the heat transfer coefficient is not an observed effect of the hydraulic diameter. However, the widths and aspect ratio of each rectangular channel influence the heat transfer coefficients. Table 1 shows the details of the tube dimension. The condensation heat transfer coefficients of refrigerant inside a rectangular tube are dependent on the surface tension and the shear stress interface of vapor and liquid phases. The condensed liquid film at the top and bottom of the channel is thinner as the width of the rectangular channel increases. These results show that due to surface tension, the condensed liquid drainage to the edges generated a thin liquid film.

4. Conclusions

The condensation heat transfer coefficients of R1234yf inside two types of multiport minichannel tubes were investigated experimentally over a large range of test conditions. The installation and data reduction were described and defined; the results were analyzed and discussed. The condensation heat transfer coefficient increases with increasing vapor quality, mass flux, and heat flux. The effects of cross-section geometry on the condensation heat transfer characteristic are dependent on the surface tension and vapor shear stress.

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